The OCTOBER, 1931

OCT 9 - 1931

Heating · Piping and Air Conditioning

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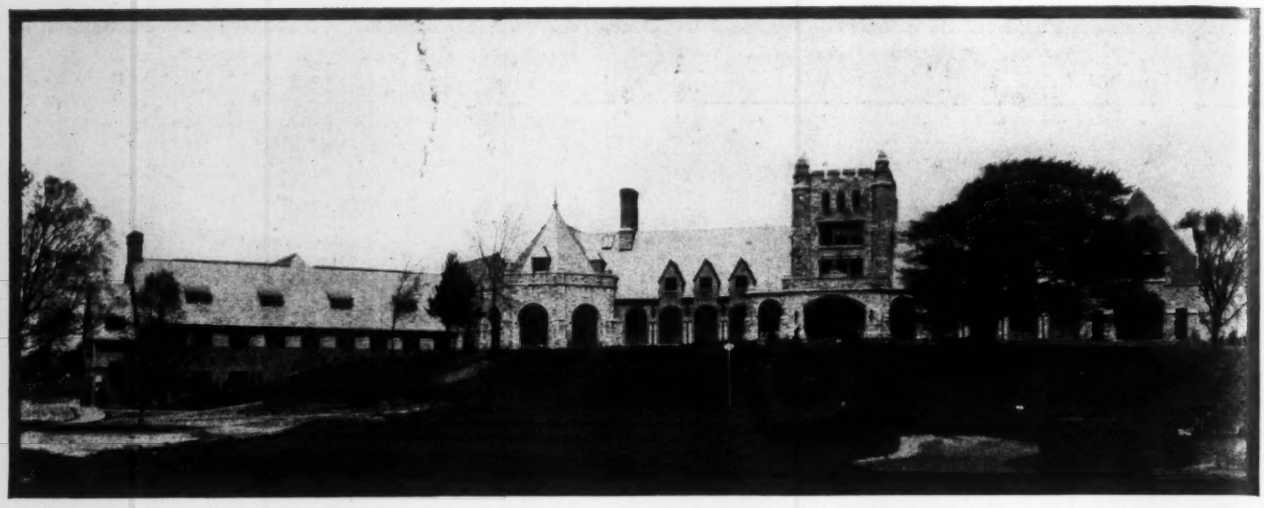
UNIT CONDITIONERS SUPPLEMENT CENTRAL SYSTEMS

Journal Section of A. S. H. V. E.



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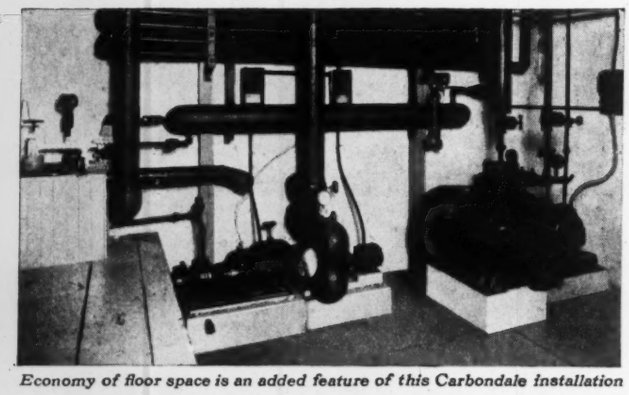

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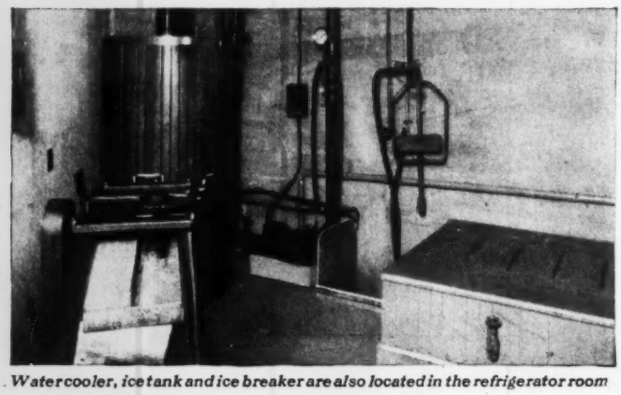

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The EDITOR'S PAGE

Heating · Piping and Air Conditioning

FROM ARTICLES IN THIS ISSUE

Heating Records

"While this method of recording steam consumption was originally devised for office building use, it is felt that it should be applicable to industrial plants and other types of structures as well."

Vacuum Cleaner Piping

"To secure permanent efficiency (in central vacuum cleaning systems), piping of ample size with outlets at easily accessible points is quite necessary. The piping should not be so small as to require too high a velocity with consequent loss in vacuum as well as possible sand-blast wear. On the other hand, it is essential that the piping should not be excessive in size, as this means a reduction in the air velocity which results in dust settling in the horizontal pipe lines connecting all risers."—*Lester C. Smith.*

Vibrating Machinery

"Since mechanical vibrations are readily transmitted through the solid structure of a building, it is extremely important that all mechanical equipment used in air conditioning, in which vibrations are generated, be thoroughly insulated from the solid structure of the building."—*Vern O. Knudsen.*

Air Inlets in Schools

"A feature of the school lies in the fact that all air inlets to the rooms are along the floor against the exterior walls. The radiators in the classrooms are placed above these air supply openings, so that the entering air rises over and around the radiators, increasing their output."—*Samuel R. Lewis.*

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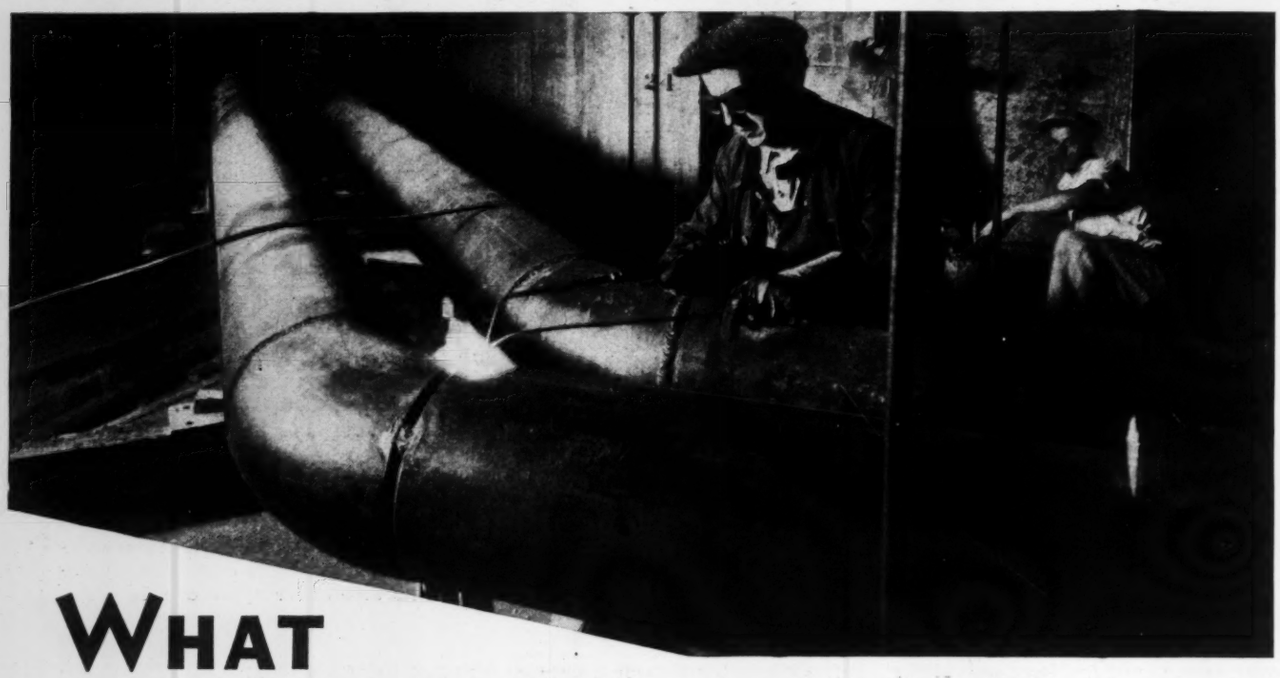

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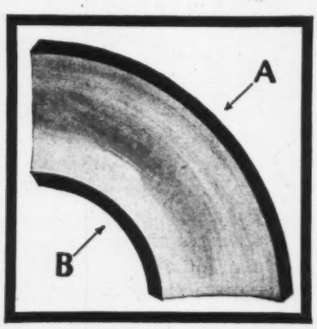

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NOTE: *Unretouched photograph of a stock Tube-Turn cut away to show uniform walls. Walls painted black for sharper photographing.*

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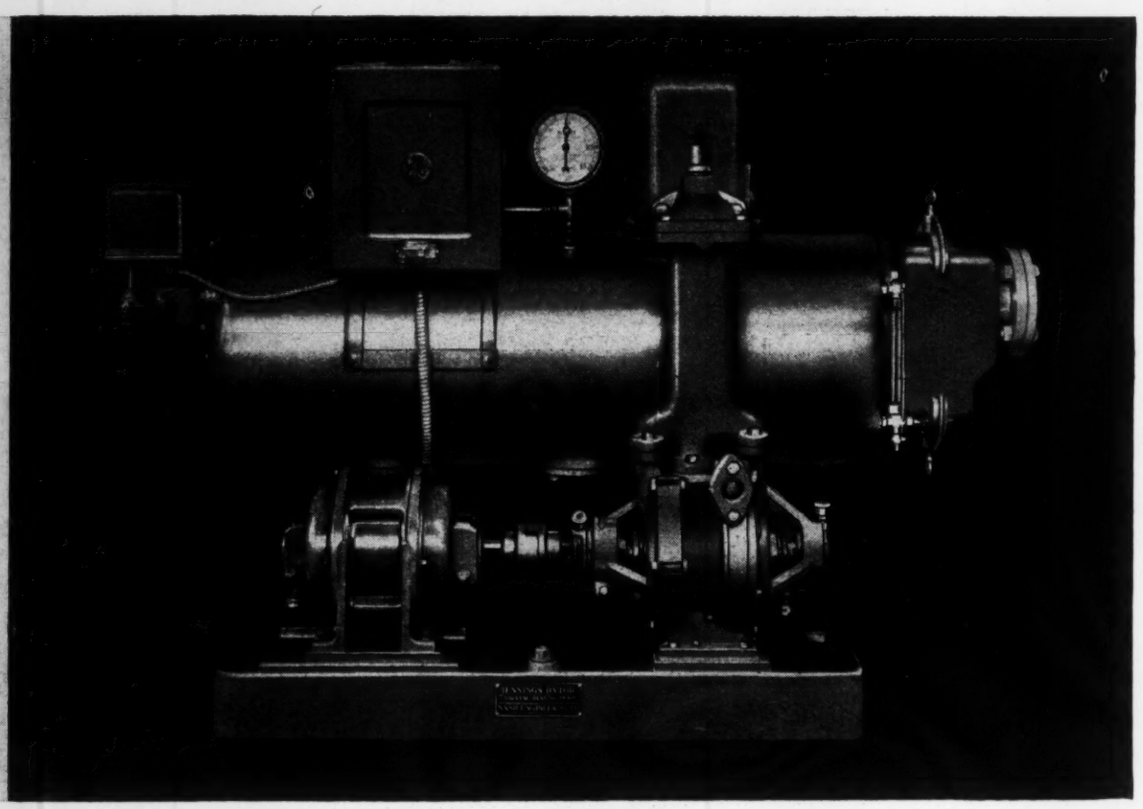

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Heating · Piping and Air Conditioning

Vol. III October, 1931 No. 10

How Sound Is Controlled

By Vern O. Knudsen*

FUTURE developments in air conditioning and acoustics are most intimately related. Office buildings, industrial buildings, hospitals, hotels, and apartment houses require not only properly conditioned air, but also the adequate suppression of noise. The elimination of noise in buildings may require heavy doors and windows (perhaps double glazed or double sashed windows), or it may be necessary to dispense with the window openings altogether. The elimination of external noise in buildings, therefore, requires the installation of never-failing equipment for the supply of properly conditioned air. The transmission of noise through window openings, and from room to room by way of the windows, is at present the limiting factor in the reduction of noise in buildings, and the adequate suppression of noise in buildings can not be successfully attained if it is necessary to have open windows for the purpose of ventilation.

Reduction of External Noise Focusses Attention on Ventilating Noise

It is obvious that the reduction of external noises in buildings will tend to focus attention upon noises which originate in air conditioning equipment, since the suppression of other disturbances will make the equipment noises much more conspicuous. Thus, the noise from a ventilating system may not be noticed in an ordinary theater owing to the presence of other noises which mask the noise from the ventilating

The acoustical problems which arise in connection with air conditioning are: (1) the selection of equipment which operates without generating excessive noises; (2) the location and insulation of the equipment room so that no direct noises are transmitted through its walls and ceiling into adjacent rooms; (3 the mounting of equipment on flexible supports which have been DESIGNED to provide sufficient insulation to prevent solid-borne vibrations from disturbing any part of the building; (4) the design of suitable filters for the ducts to prevent sound transmission; and (5) the control of humidity to provide relatively humid air for speech and music rooms and relatively dry air in office and work rooms.

VENTILATING FAN AND MOTOR MOUNTED ON RESILIENT STEEL CHAIRS

* University of California at Los Angeles.

equipment, whereas in a sound stage, such as is used in the making of talking pictures, all other external noises have been insulated so that the noise from the ventilating equipment may seem excessively loud. It is apparent that the elimination of noises in buildings will require proper control of the noises attributable to air conditioning equipment.

Even at the present time there are installations of ventilating equipment in auditoriums the noise of which constitutes the greatest acoustical defect in the auditorium. The writer was recently consulted with regard to the acoustical defects in a certain high school audi-

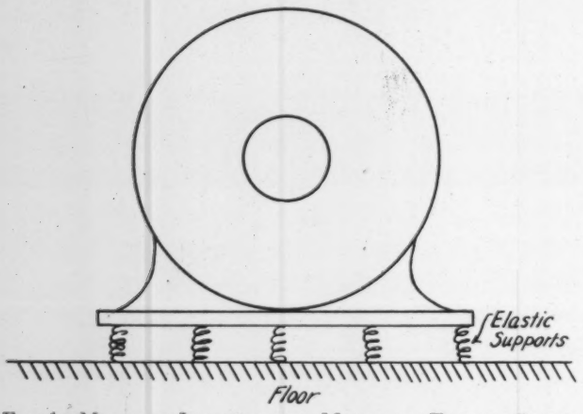


FIG. 1—MACHINE INSULATED BY MEANS OF ELASTIC PAD OR SUPPORTS. THE NATURAL FREQUENCY OF THE MASS m ON ITS ELASTIC SUPPORTS SHOULD BE LOW IN COMPARISON WITH THE VIBRATIONAL FREQUENCIES GENERATED BY THE MACHINE; OTHERWISE, THE ELASTIC SUPPORTS MAY BE WORTHLESS

torium, and found that the chief defect was caused by the noise from the ventilating fan and motor. Speech tests† conducted in the empty auditorium with the fan and motor in operation showed an average speech articulation of 66 per cent.

(This means that out of every 100 meaningless speech sounds that were called out by a speaker in the auditorium, only 66 were heard correctly.)

When the same tests were repeated with the fan and motor shut down the percentage articulation increased to 80 per cent. If the tests had been conducted with an audience present the percentage articulation would have been slightly higher in both tests, owing to the absorption supplied by the audience, but the interfering effect of the noise would have been fully as inimical to the hearing of speech with an audience present as it was with no audience present. Since an articulation of at least 75 per cent is required for satisfactory hearing, it is obvious that the noise from the ventilating equipment in this auditorium was responsible for a very troublesome acoustical defect.

In the present article the writer proposes to consider the problems which arise in connection with the control of sound in relation to the air conditioning of rooms. The proper design and installation of this equipment will facilitate noiseless operation without sacrificing the efficiency of the equipment. Specifically, such problems as the insulation of solid-borne noise, the insulation of noise originating in the equipment room, the prevention of noise transmission through the ducts, the control of air currents in auditoriums, and the relation of humidity to room acoustics, will be considered in some detail, and practical methods of solving the problems which arise in typical installations will be described.

The Insulation of Solid-Borne Noise

Since mechanical vibrations are readily transmitted through the solid structure of a building, it is extremely important that all mechanical equipment used in air conditioning, in which vibrations are generated, be thoroughly insulated from the solid structure of the building. An almost universal notion prevails that the vibrations generated by machinery can be insulated from a building simply by placing a slab of cork or a layer of hair felt between the machinery and the floor of the room. If the machinery is sufficiently heavy, and the cork or felt sufficiently resilient, this expedient may suffice. On the other hand, if the machinery is not sufficiently heavy to "load" the cork or felt support to the extent that the natural frequency of the machinery on the cork or felt is low in comparison with the frequency generated by the equipment, the cork or felt may be of little avail. The insulation of vibration can be accomplished by means of suitable elastic supports or suspensions, but the design of these elastic supports should be based upon calculation rather than guesswork.

The theory of the insulation of vibration was first worked out by Soderberg.[1] Suppose a machine of mass m be supported by an elastic pad as shown in Fig. 1. The amount of vibratory force communicated by the machine to the floor or foundation upon which it rests will be determined by the elastic and viscous properties of the pad. The ratio of the vibratory force communicated to the floor or foundation with the machine resting upon the pad and with the machine resting directly upon the floor is given by

$$\tau = \sqrt{\frac{r^2 + \dfrac{1}{4\pi^2 n^2 c}}{r^2 + \left(2\pi nm - \dfrac{1}{2\pi nc}\right)^2}} \qquad (1)$$

where τ is the so called "transmissibility" of the support, c is the compliance (that is the reciprocal of the force constant), r is the mechanical resistance owing to the viscous forces within the support, and n is the frequency of vibration generated by the machine which is to be insulated—such as the commutation frequency of a motor or the blade frequency of a fan. A plot of τ as a function of frequency is shown in Fig. 2. If the pad is to be of any value in the prevention of solid-borne vibrations, the value of τ must be considerably smaller than unity. If the fundamental frequency of vibration generated by the machine happens to coincide with the natural frequency of the mass of the machine resting on the elastic pad, a condition of resonance will be estab-

† Speech articulation tests are used by telephone engineers to test the speech transmission properties of telephone equipment. One person calls out meaningless monosyllabic speech sounds into a transmitter, and an observer listening with a telephone receiver writes down what he hears. By comparing his recorded list with the called one the number and nature of the errors in the recognition of speech can be determined. Similar tests can be conducted in auditoriums to determine the effects of noise or reverberation upon the hearing of speech. By conducting speech tests first in the quiet and then in the presence of a noise, such as may be produced by a ventilating fan, it is possible to determine the damage which noise does to the recognition of speech. (See V. O. Knudsen, The Hearing of Speech in Auditoriums, *Journal* of the *Acoustical Society of America*, October, 1929.).

1 C. R. Soderberg, *The Electric Journal*, January, 1924, and succeeding articles. See also V. O. Knudsen, *Phys. Rev.*, 32, 324, 1928, and A. L. Kimball, *Jour. Acous. Soc. Amer.*, 2, 297, 1930.

lished and the machine will exert a greater force upon the foundation than it would if the pad were completely removed. This is shown clearly in Fig. 2 where the resonance peak corresponds to the natural frequency of the machine mounted on the elastic support. It is necessary therefore that the elastic support be sufficiently compliant, and the mass of the machine sufficiently heavy, that the natural frequency of the mass m upon its elastic support will be low in comparison with the frequencies which are generated by the machine. Thus, if the principal vibrations in the machine be of the order of 100 vibrations per second, the natural frequency of the machine mounted on its elastic support should not exceed about 20 vibrations per second.

If a slab of insulating material be placed under the entire foundation of a machine, as is often done in practice, it may happen that the natural frequency of the machine on its elastic support will be nearly the same as the frequencies which are to be insulated, in which case the elastic support will be worse than nothing. In general, as Equation 1 shows, both m and c should be as large as possible if the vibrations of the machine are to be effectively insulated from the solid structure of the building. Further, the machine should rest upon a very rigid floor so that the elastic yielding of the floor is prevented from communicating the machinery vibrations to the solid structure of the building.

The elastic support under the machine acts as a low-pass filter which passes all frequencies below about two times the natural frequency of the machine mounted on its elastic support, but prevents all frequencies above about $\frac{1}{\pi\sqrt{mc}}$ from reaching the solid structure of the building. The principal influence of the internal mechanical resistance r is to limit the vibration at the resonant frequency. It is generally advisable therefore to use materials which have an appreciable internal resistance.

The values of c and r can be determined for any specimen of flexible material and, when known, can be used to determine the insulation value of any particular set-up. The value of c can be obtained by making static measurements of the amount of displacement of the compressed support for each additional unit of the compressing force. If this be done for a specimen of the flexible material of a certain thickness and area of cross-section, the compliance can be determined for any other thickness or area from the relation that c will be directly proportional to the thickness and inversely proportional to the area of the flexible support. When the internal resistance r is not too large, it can be determined by observing the successive amplitudes of the free vibrations of a mass m which rests upon a specimen of the flexible material, and solving for r by the usual log decrement method. Or, if the damping be so great that the free motion of m is non-oscillatory, r can be obtained from measurements on the experimentally determined resonance curve of the forced vibrations of m, or from measurements of the rate of return of m when it is given an initial displacement.

This last method, which may be regarded as a static method, is similar to the electrical method of determining the resistance R in series with an inductance L and a capacitance C by discharging the condenser through the inductance and resistance, and obtaining the "decay" curve. If the resistance of a certain specimen of material, as cork, felt, or rubber, has been determined by any of these methods, the resistance for any other thickness or area of the material can be appropriately determined from the relation that the resistance will be inversely proportional to the thickness and directly proportional to the area of cross section of the flexible support. Thus, if the values of c and r for a flexible material be known, it is possible to calculate, by means of Equation 1, the amount of insulation that will be obtained from the use of this material as a flexible support for a piece of equipment having a mass m. Tables are also available.

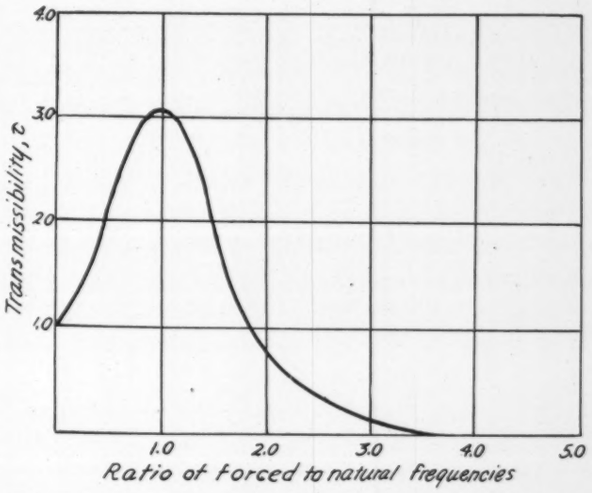


FIG. 2—SHOWING HOW THE TRANSMISSIBILITY OF AN ELASTIC SUPPORT, SUCH AS IS SHOWN IN FIG. 1, DEPENDS UPON FREQUENCY. IF THE NATURAL FREQUENCY OF THE MASS m ON ITS ELASTIC SUPPORT BE ABOUT THE SAME AS THE FORCED FREQUENCIES (SUCH AS COMMUTATOR OR FAN BLADE FREQUENCES) THE "INSULATING" SUPPORT WILL ACTUALLY FACILITATE RATHER THAN IMPEDE THE TRANSMISSION OF MACHINERY VIBRATION TO THE FLOOR OF THE ROOM. BUT IF THE FREQUENCIES OF THE VIBRATIONS GENERATED BY THE MACHINERY BE FOUR OR FIVE TIMES THE NATURAL FREQUENCY OF THE MASS m ON ITS ELASTIC SUPPORT, THERE WILL BE VERY LITTLE VIBRATION TRANSMITTED TO THE FLOOR

In general, there are two principal points to observe in the design of a flexible support for any particular piece of equipment; namely, the material should have a relatively large compliance c, and *it should be loaded to nearly the upper safe limit* of loading. In recent years there have been developed a number of flexible metallic supports which are finding application.

A Typical Example

In order to show a specific example of sound insulation, suppose we have a machine weighing 1,000 lb. and having a base area of 20 sq. ft. Suppose further that the principal vibration of the machine has a frequency of 100 cycles per second (most machinery vibrations are less than 150 vibrations per second, and the assumed frequency of 100 is quite representative of typical machines). Suppose, as is frequently done, that a 1-in. slab of corkboard, 1.10 lb. per board ft., be placed between the machine and the floor. The loading on the cork will then be only 50 lb. per sq. ft., or slightly more

than 1/3 lb. per sq. in. (It is assumed that the compliance c in cm./dyne for a specimen 1-in. thick and I sq. cm. in cross-section is 0.25×10^{-6} and the resistance r in mechanical ohms is 0.15×10^{5}.)

The calculation of the "transmissibility" can be done in the following manner:

Mass of machine in grams $= 1000 \times 454 = 4.54 \times 10^{5}$.

Area of base in sq. cm. $= 20 \times 144 \times 2.54 \times 2.54 = 1.86 \times 10^{4}$.

Therefore, the compliance of entire support, 1 in. thick and 20 sq. ft. in cross section, is $.25 \times 10^{-6} \times \frac{1}{1.86 \times 10^{4}} = .134 \times 10^{-10}$ cm./dyne, and the resistance of the entire support is $.15 \times 10^{5} \times 1.86 \times 10^{4} = 0.28 \times 10^{9}$ mechanical ohms. Therefore

$$\tau = \sqrt{\frac{(.28 \times 10^{9})^{2} + \frac{10^{20}}{4\pi^{2} \times \overline{100^{2}} \times \overline{0.134^{2}}}}{(.28 + 10^{7})^{2} + \left(2\pi \times 100 \times 4.54 \times 10^{5} - \frac{10^{10}}{2\pi \times 100 \times 0.134}\right)^{2}}} = 0.93$$

Consequently, it is seen that the "transmissibility" is nearly equal to unity, and that the support therefore is not satisfactory for insulating vibrations of 100 or fewer per second.

If now the amount of cork be reduced so that it is loaded to 10 lb. per sq. in., the total area of the supporting cork will be only 100 sq. in., or 645 sq. cm. The compliance of the entire support will now be $0.25 \times 10^{-6} \times \frac{1}{645} = 0.39 \times 10^{-9}$ cm./dyne, and the resistance will be $0.15 \times 10^{5} \times 645 = .97 \times 10^{7}$ mechanical ohms. Therefore

$$\tau = \sqrt{\frac{(.97 \times 10^{7})^{2} + \frac{10^{18}}{4\pi^{2} \times \overline{100^{2}} \times \overline{0.39^{2}}}}{(.97 \times 10^{7})^{2} + \left(2\pi \times 100 \times 4.54 \times 10^{5} - \frac{10^{9}}{2\pi \times 100 \times 0.39}\right)^{2}}} = .037$$

It is seen, therefore, that with the bearing surface on the cork reduced to 100 sq. in. (that is, with the cork loaded to 10 lb. per sq. in.) the "transmissibility" is reduced to 0.037, or the amplitude of vibration transmitted to the floor will be only about 1/27 of what it would be if the machine were mounted directly upon the floor. These two numerical examples will serve to show not only the manner of making the calculations, but also the importance of selecting the proper type and design of flexible supports for insulating the vibrations of a machine from the rigid structure of a building.

Insulation of Noise Originating in the Equipment Room

In this section we shall deal particularly with the insulation of the air-borne noise which is generated within the equipment room. In many rooms this direct noise may reach a level as high as 60 or 70 db.[2], in which case it is necessary that the walls and ceiling of the equipment room provide a sufficient amount of sound insulation to prevent this noise from being transmitted through the boundaries of the room into adjacent rooms.

In the first place, everything possible should be done to minimize the noise *at its source*. This is a mechanical problem in the design and selection of the equipment. Secondly, the amount of noise generated in the room can be reduced approximately 10 db. by lining the walls and ceiling of the equipment room with a highly absorptive material, such as a mineral wool blanket, or acoustical tile. Finally, the walls and ceiling of the equipment room, together with any doors or windows in the room, should be designed so as to reduce the residual transmitted noise to such a level as can be tolerated in adjacent rooms.

Thus, suppose that the amount of noise in a room after the walls have been lined with absorptive material is 55 db., and that the amount of noise which can be tolerated in the adjacent room is 10 db. It then will be necessary to design walls and ceiling for the room which will provide an insulation of at least 45 db. This will require walls and ceiling having an insulation the equivalent of a 6-in. solid brick wall. There are many types of construction which will provide this amount of insulation, such as reinforced concrete, brick or stone and mortar, or staggered stud partitions with a layer of insulation suspended between the double walls.

Noise Through Ducts

The most troublesome sources of noise from air con-

[2] Abbreviation for decibel. The decibel is a commonly used unit for the intensity of noise or sound. A noise of 60 db. is a noise which has an intensity one million times greater than a noise which is just barely audible; a noise of 70 db. has an intensity ten million times greater than a noise which is just barely audible.

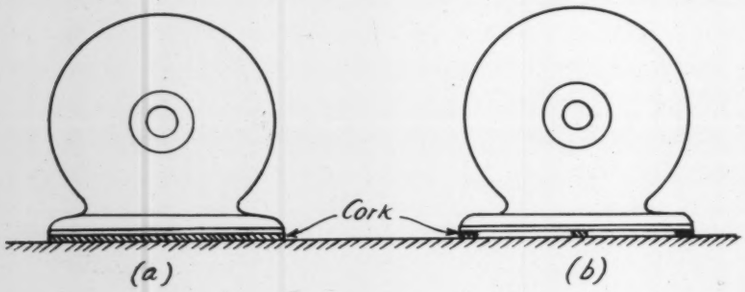
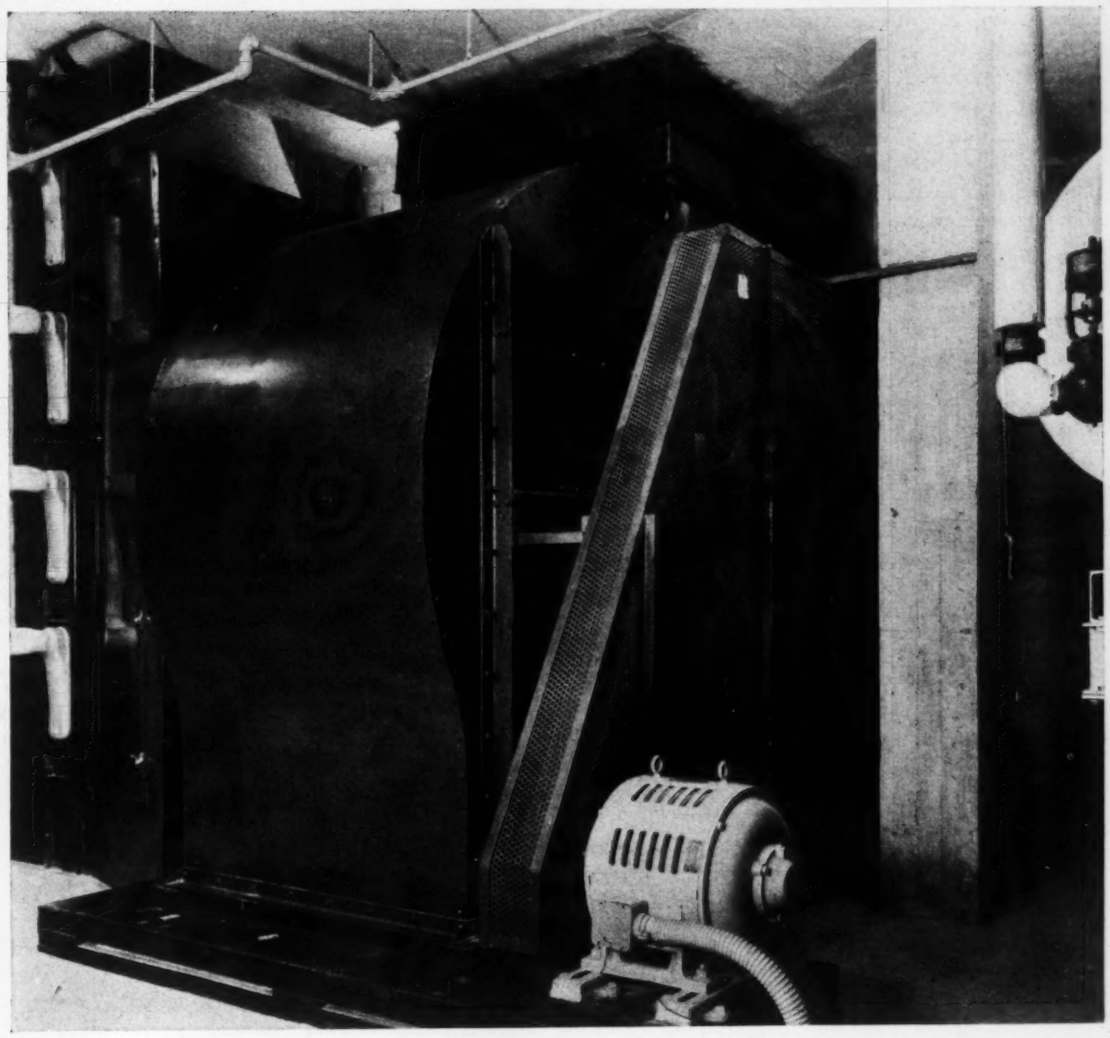


FIG. 3—THE MOUNTING OF A SLAB OF CORK UNDER A FAN OR MOTOR, AS SHOWN IN (a), MAY RESULT IN A CONDITION IN WHICH THE NATURAL FREQUENCY OF THE MASS OF THE MACHINE ON ITS CORK SUPPORT SYNCHRONIZES WITH THE FUNDAMENTAL FREQUENCY OF VIBRATION OF THE MACHINE, IN WHICH CASE THE CORK SUPPORT WILL ACCENTUATE RATHER THAN RELIEVE THE VIBRATION COMMUNICATED TO THE FLOOR OR FOUNDATION. IF THE BEARING SURFACE ON THE CORK BE GREATLY REDUCED, AS SHOWN IN (b), THE NATURAL FREQUENCY OF THE MASS OF THE MACHINE ON ITS CORK SUPPORT WILL BE MUCH LOWER THAN THE FREQUENCY OF VIBRATION OF THE MACHINE, AND THE SUPPORT WILL THEN RELIEVE THE VIBRATION COMMUNICATED TO THE FLOOR



Fan and Motor in the Los Angeles Theater, Los Angeles, California. Note That the Base of the Fan and Motor Is Mounted Upon Strips of Insulation Cork, Making the Natural Frequency of the "Load" on the Cork Low in Comparison with the Mechanical Frequencies Produced by the Operation of the Fan and Motor. Note Also the Canvas Connection Between the Fan and the Duct System

ditioning equipment are fan and motor noises which are transmitted through the ventilating ducts. Tests which have been made at the Bureau of Standards and elsewhere on the transmission of sound through speaking tubes indicate that the amount of sound which is transmitted through a tube or duct depends upon the length of the duct, the cross-sectional area of the duct, and the material out of which the duct is made. The reduction of sound transmitted through such a tube or duct, expressed in decibels, is proportional to the length of the duct, is inversely proportional to the 0.76 power of the diameter of a circular tube,[3] and is proportional approximately to the coefficient of sound absorption of the material which comprises the interior surfaces of the tube or duct. It is apparent therefore that long, narrow ducts, lined with material which is highly absorptive for sound, will provide a high degree of insulation for sounds which are transmitted through ducts. By introducing suitable sections of absorptive filters between the ventilating fan and the outlet, or between the outlets for adjacent rooms, it is possible to eliminate all of the difficulties which arise in connection with the transmission of noise through ventilating ducts. The problem is an engineering one which can be worked out prior to the installing of the equipment, and it can be calculated in such a way as to meet satisfactorily rigorous demands for silent operation.

[3] "Transmission of Sound through Voice Tubes," Technologic Papers of the Bureau of Standards, No. 333, 1926.

Air Currents

In general, the motion of air resulting from the ventilating of rooms is not sufficient to introduce any appreciable difficulty in auditoriums, except where noise may originate from the issuing of high speed air from nozzles. However, by proper stream-lining of the nozzles, it is possible to work with speeds which are adequate for all practical purposes without producing any disturbing noises. Since sound is propagated with a velocity of more than 1,100 feet per second, the velocity of the air would have to attain speeds of at least 20 to 30 feet per second before these wind velocities would have any appreciable influence upon the propagation of sound.

If there is to be any appreciable motion of air in an auditorium, it is slightly advantageous to have the upper

layers of air moving in a direction from the stage toward the audience, as this will tend to refract the sound waves down toward the audience. However, unless the speed of the air is as great as 20 or 30 feet per second the amount of refraction will not be appreciable. It may be stated, therefore, that as a rule the motion of air in an auditorium does not have an appreciable effect upon the acoustical properties of the room.

The Effect of Humidity Upon the Acoustics of an Auditorium

Recent experiments[4] have shown that the humidity of the air has a rather marked influence upon the rate of absorption of high pitched sounds in a room. High frequency sounds are propagated better in humid air than in dry air, and since high frequency sounds are particularly important for the preservation of good quality in speech and music it is advantageous to maintain the air in a room at a relatively high humidity—not less than about 55 or 60 per cent. On the other hand, where it is desirable to absorb all frequency components of sound, as for the reduction of noise in offices, it is advantageous to maintain a relatively dry air.

The time of reverberation in a room is given by

$$t = \frac{0.049\ V}{-S \log_e (1-\alpha) + 4mV}, \qquad (2)$$

where V = volume of room in cu. ft.; S = interior surface of room; α = average coefficient of sound-absorption of the interior surface of the room; and m is the absorption coefficient of the air in the room.

The coefficient m depends upon the frequency of the sound and the humidity of the air. At a temperature of 70 F, and for sound waves having a frequency of 4,096 vibrations per second, m = 0.0027 at 20 per cent relative humidity, 0.0018 at 50 per cent R. H., and 0.0013 at 80 per cent R. H. It will be seen therefore that the absorption of sound in the air is twice as great at a relative humidity of 20 per cent as it is at a relative humidity of 80 per cent. (This explains why sounds in the open travel so much better on humid days than they do on dry days.) Although this dependence of absorption upon humidity is characteristic of low frequency as well as high frequency sound, the actual amount of absorption in the air is negligible for frequencies below about 1,024 vibrations per second. However, the absorption of the higher frequencies in the air is a significant factor, and its dependence upon humidity calls for careful consideration in planning the air conditioning equipment for buildings.

[4] V. O. Knudsen, "The Effect of Humidity upon the Absorption of Sound in a Room," *Jour. Acous. Soc. Amer.*, July, 1931.

Control of Sound Not Guess-work

In conclusion, the acoustical problems which arise in connection with the air conditioning of buildings are: (1) the selection of equipment which operates without generating excessive noises; (2) the location and insulation of the equipment room so that no direct noises are transmitted through its walls and ceiling into adjacent rooms; (3) the mounting of the equipment on flexible supports which have been designed (based upon calculations) to provide sufficient insulation to prevent solid-borne vibrations from disturbing any part of the building; (4) the design of suitable filters for the ducts to prevent sound transmission from the equipment room to audience or work rooms, or from room to room; and (5) the control of the humidity so as to provide relatively humid air in speech and music rooms, and relatively dry air in offices and work rooms. All of these problems can be solved quantitatively in advance of installing the equipment, so that the control of sound in the air conditioning of buildings can be removed from the realm of guess-work and empiricism, and placed upon an unfailing and rigorous engineering basis.

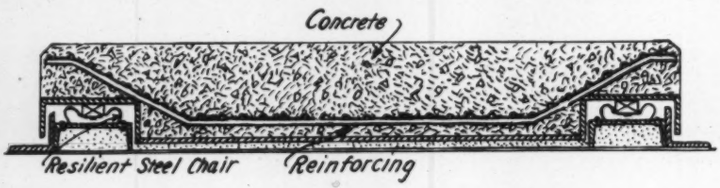


FIG. 4—SECTION OF SOUND INSULATION MACHINE BASE WITH ALL STEEL FRAME AND CONCRETE DECK

A. S. T. M. Committee Approves Piping Standards

Committee E-10 on standards of the *American Society for Testing Materials* at its recent meeting in Philadelphia approved thirteen new tentative specifications and methods of test and the revision of four existing tentative standards, according to a bulletin received from R. E. Hess, assistant secretary of the *A. S. T. M.* The titles and designations of the five relating to piping are as follows:

New Tentative Standards Submitted by Committee A-1 on Steel—

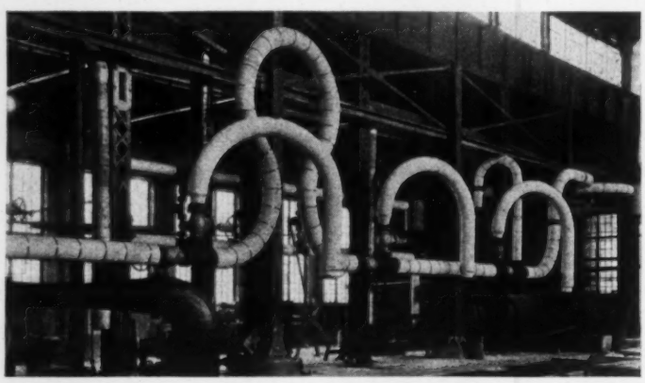
Specifications for Electric-Fusion-Welded Steel Pipe (*A* 134-31 *T*).

Specifications for Electric-Resistance-Welded Steel Pipe (*A* 135-31 *T*).

Specifications for Forge-Welded Steel Pipe (*A* 136-31 *T*).

Specifications for Lock-Bar Steel Pipe (*A* 137-31 *T*).

Specifications for Riveted Steel and Wrought-Iron Pipe (*A* 138-31 *T*).



Records Show How to Save Heating Steam

Here's a method of keeping records of heating steam or fuel consumption which has been tried and found practical. It will point the way to savings in heating costs in industrial plants, office buildings and other structures and it's not too complicated to use.

APPROXIMATELY forty office buildings in Chicago kept heating records during the winter of 1930-31 according to the system devised by the committee on plant operations of the *Building Managers' Association of Chicago*. These data are now being analyzed, but as there are so many conditions which affect each building's records, it is unknown at the present time just what conclusions can be drawn.

However, as an aid to the operating engineer, it is felt that this system is invaluable, as it will enable each engineer to know accurately his heating steam consumption and will point the way to economies in operation. It is reported that one large building made savings in steam amounting to several hundred dollars per month through the use of these charts; the records are being studied with care by the committee in order to verify these results.

While this method of recording steam consumption was originally devised for office building use, it is felt that it should be applicable to industrial plants and other types of structures as well.

How to Use the Charts

In the operation of a heating system, actual records of the cost and knowledge of the factors making up the cost of heat are extremely important. Nothing can be operated intelligently without a definite knowledge of performance. Only by continual checking and persistent endeavor on the part of the operator can the costs be reduced to a minimum. Simple forms for keeping

BUILDING PROSPERITY — MONTH OF JANUARY, 1930.

DAY		POUNDS OF STEAM OR FUEL				HEATING HOURS			OUTSIDE TEMPERATURE			Degree	Lbs. Steam or Fuel per Degree Day	Lbs. Steam or Fuel per Degree Day per 1000 cu. ft. Building
Of Month	Of Week	8 A.M.-4 P.M.	4 P.M.-12 MDT.	12 MDT.-8 A.M.	TOTAL 24 HOURS	Steam On At	Steam Off At	TOTAL HOURS	Mean	Normal	Depart. from Normal	Days		
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
1	WED.	32,000		480	32,480	10:00AM	3:00PM	5.0	41°	25°	+16°	24	1,353	.1353
2	THURS.	115,200	19,200	28,320	162,720	6:15AM	6:00PM	11.75	36°	25°	+11°	29	5,611	.5611
3	FRI.	136,800	41,760	32,160	210,720	6:15AM	7:45PM	13.5	27°	25°	+2°	38	5,545	.5545
4														
5														
6														
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30														
31														
TOTALS														
AVERAGE														

Columns (10) and (11) obtained from U. S. Weather Bureau
Column (12) -- Column (10) minus Column (11)
Column (13) -- 65° minus Column (10)

Column (14) Divide Column (6) by Column (13)
Column (15) Multiply Column (14) x 1000 and Divide by volume of building.

FIG. 1—THE DAILY HEATING RECORD IS APPLICABLE TO ANY BUILDING FOR ANY MONTH IN ANY CITY. SIMPLICITY IS THE KEYNOTE OF THIS RECORD-KEEPING SYSTEM

records enable both the engineer and the building manager to have an intelligent indication of the operation and costs. The daily heating record shown in Fig. 1 has been prepared to provide such a record and is applicable to any building for any month in any city.

Column (1) lists the days of the month and column (2) is filled in with the corresponding days of the week in the same month. This latter explains variations in daily heat consumption due to Saturdays, Sundays and holidays.

Columns (3), (4), (5) and (6) are filled in each day to indicate the actual pounds of steam or fuel for that day. Column (7) shows the time steam is turned on, column (8) when it is shut off, and column (9) indicates the number of hours during which heat is supplied during the day.

In column (10) is posted each day the mean temperature, which is half-way between the maximum and minimum outside temperature. This mean and normal in column (11) are obtained from a list sent out daily by the local office of the U. S. Government Weather Bureau to subscribers at a cost of $2.00 per year. This mean temperature is especially important to know because the heat required for a building varies with variations in the outside temperature, of course.

In column (13) are noted the degree days. This is the difference between the outside mean temperature and 65 F. In other words, if the mean temperature is 64 F the degree days would be 65 minus 64, or 1. If the mean temperature were 50 there would be 65 minus 50, or 15 degree days, etc. The upper temperature of 65 F is taken because on those days when the mean is 65 or above, no heat will be required in the building. This is because 65 is half-way between 60 F minimum during the night and 70 F in the daytime when heat would be used, if at all. Due to the higher daytime outside temperature it would be unnecessary to turn on the heat.

This item of degree days is the basis of comparison shown in column (14), which item is obtained by dividing the total pounds of steam or fuel, column (6), by the degree days in column (13).

There is still another item necessary before the steam consumption of two buildings may be compared. This is the size of the building in cubic feet. For the purposes of comparison on a uniform basis, the cubic foot volume of each building should be measured in accordance with the practice generally recog-

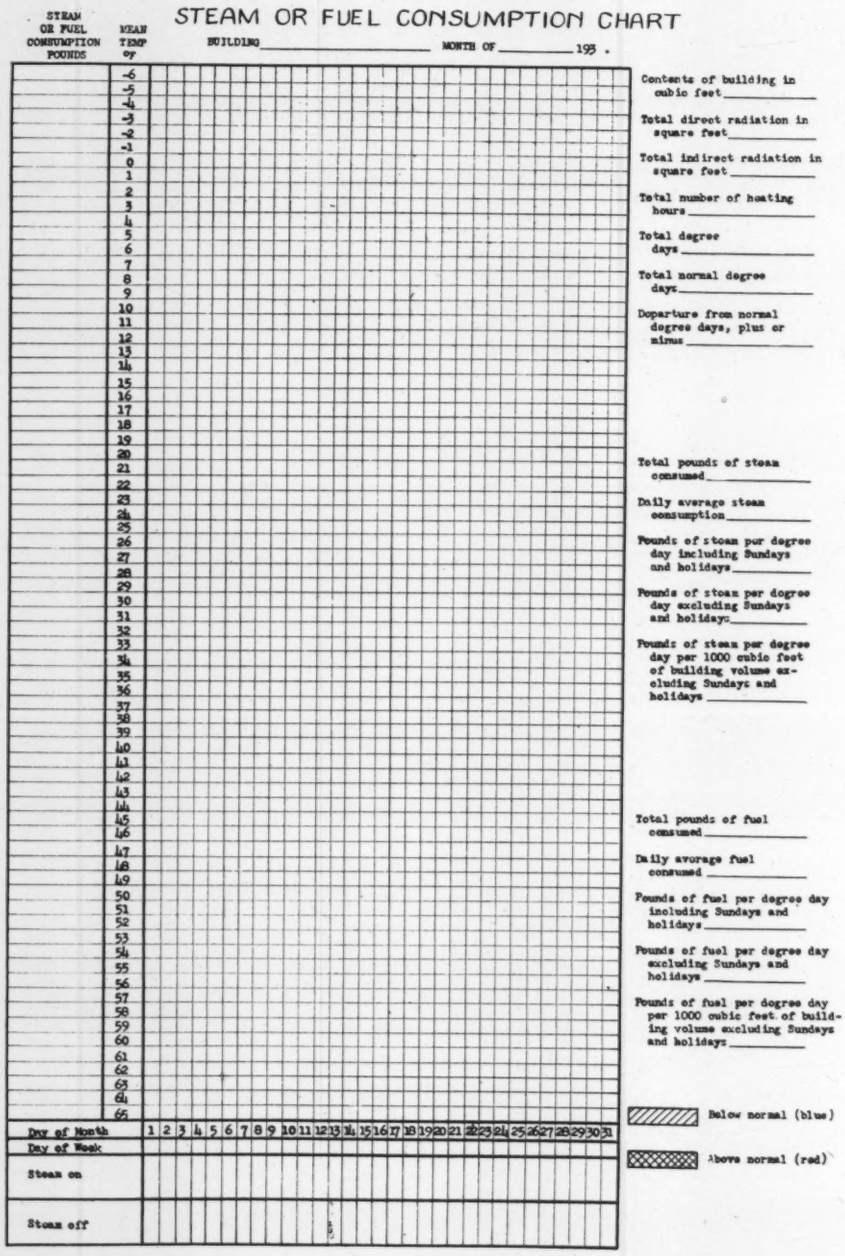
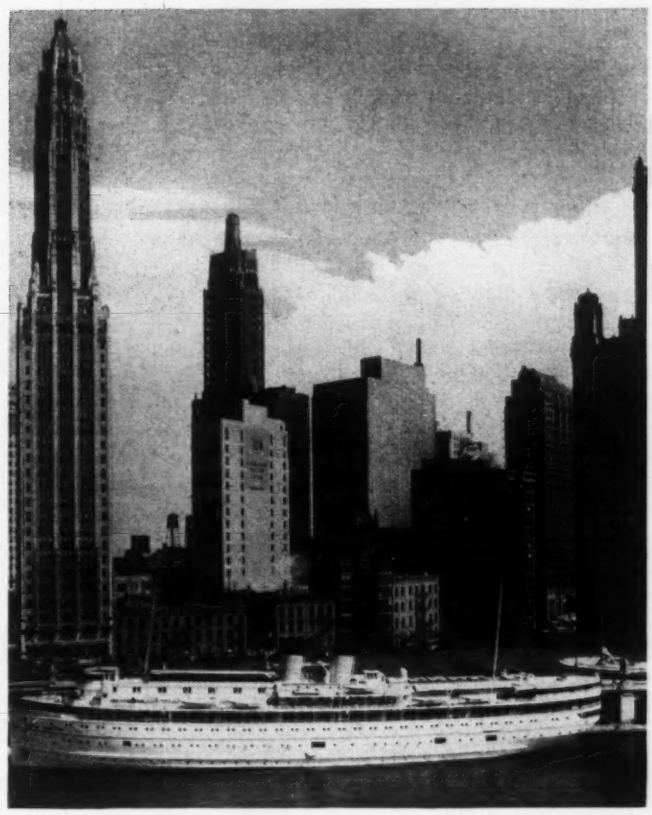

STEAM OR FUEL CONSUMPTION CHART
BUILDING
MONTH OF
19
STEAM OR FUEL CONSUMPTION POUNDS
MEAN TEMP °F
Contents of building in cubic feet
Total direct radiation in square feet
Total indirect radiation in square feet
Total number of heating hours
Total degree days
Total normal degree days
Departure from normal degree days, plus or minus
Total pounds of steam consumed
Daily average steam consumption
Pounds of steam per degree day including Sundays and holidays
Pounds of steam per degree day excluding Sundays and holidays
Pounds of steam per degree day per 1000 cubic feet of building volume excluding Sundays and holidays
Total pounds of fuel consumed
Daily average fuel consumed
Pounds of fuel per degree day including Sundays and holidays
Pounds of fuel per degree day excluding Sundays and holidays
Pounds of fuel per degree day per 1000 cubic feet of building volume excluding Sundays and holidays
Below normal (blue)
Above normal (red)
Day of Month
Day of Week
Steam on
Steam off

FIG. 2—THE STEAM OR FUEL CONSUMPTION CHART IS PLOTTED FROM THE DATA ON THE DAILY HEATING RECORD. IT IS AN EASILY READ METHOD OF KEEPING A VISUAL INDICATOR OF DAILY OPERATION



SEVERAL CHICAGO OFFICE BUILDINGS

nized. These measurements begin one foot below the surface of the basement floor and include the entire structure to the average height of roof.

One thousand cubic feet are taken as the unit of volume. The pounds of steam or fuel per degree day are multiplied by 1,000 and divided by the cube foot volume of the building to obtain the pounds of steam or fuel per degree day per 1,000 cubic feet of building volume. This result is listed in column (15.) Theoretically, this figure should be constant for a building as long as the heat requirements are not changed. As an average, the figure will be about six-tenths pounds of steam per 1,000 cubic feet of building volume for a normal business day for an office building. When a higher figure is obtained, it may indicate that there is some waste or leakage in the steam used and a search should be made to find the source.

At the end of the month each of the several columns should be totaled and each total divided by the number of days in that month to find the averages for the month.

It is highly desirable that pounds of steam be used as a basis of consumption rather than fuel, inasmuch as the efficiency of the heating plant does not enter into the comparison when the steam consumption figure is used. The installation and use of such meters and instruments as are necessary to obtain an accurate record of steam consumption is recommended. Experience has demonstrated that the cost of such equipment is more than met by savings effected, in a relatively short time.

However, if any building is not equipped with sufficient meters to make an accurate determination of steam consumption, it will be found very helpful to keep a record of the pounds of fuel instead.

In order to be strictly comparable, all heating readings should be corrected by subtracting each day any steam or fuel required for hot water, restaurant or any purpose other than heating. It is assumed that the efficiency of the heating plant will remain practically constant and daily heating records by this method will furnish a day by day comparison for the engineer.

Fig. 1 shows the daily heating record, with entries made for the first three days of January, 1930. These figures are copied from the record kept in a building with a volume of approximately 10,000,000 cubic feet, in which steam consumption is recorded. These entries illustrate the difference between steam consumption on a holiday and on normal business days.

Graphical Records

A daily graph as shown in Fig. 2 is an easily read method of keeping a visible indicator of daily operation. This is plotted from the data on the log sheet shown in Fig. 1. In the left hand column is entered the estimated steam consumption for the building for a range in temperature from 65 F to 6 F below zero, the temperature being listed in the second column from the left. At the base line the steam consumption is zero to correspond with the mean temperature at 65 F. At 64 F the steam consumption will be the estimated amount required for one degree day. Where this estimated amount is not known, the use of six-tenths (0.6) pounds of steam per 1,000 cubic feet of building volume per degree day is suggested. As an example, for a building of 1,000,000 cubic feet, 600 pounds of steam would be indicated for a temperature of 64 F, 1,200 pounds for 63 F, 1,800 pounds for 62 F and so on to the top of the column.

This ratio of 0.6 pounds per 1,000 cubic feet is based on experience in Chicago office buildings and is suggested as an arbitrary beginning for those who do not know their actual steam consumption per degree day. Experience will demonstrate whether this ratio is too high or too low and in subsequent months a ratio may be used that will more closely approximate minimum requirements. In buildings not equipped with meters, where coal consumption is recorded, it is estimated that the figures in the left hand column may be one-sixth of the estimated steam consumption.

Plotting the Graph

Any economies effected will be reflected in the curves, as the chart is plotted from day to day. The daily plotting of this graph is done as follows:

First, there is plotted the horizontal line showing the actual steam consumption noted from the meter readings. Then the second horizontal line is plotted to correspond with the mean outside temperature for that day as reported by the Weather Bureau. If the second or temperature line falls higher than the first, the difference in pounds of steam between these two lines represents the amount of steam used less than the estimated required amount. If, however, the temperature line falls below the actual steam consumption line, the difference represents the amount of steam used in excess of the estimated required amount. It makes the chart more easily analyzed if this excess area is colored in red and the saved area is colored in blue. At the end of each month the chart may be carefully analyzed to see whether there is a greater amount of steam saved or used as an excess. The desirable chart shows practically an equality of red and blue. If, however, there is a large amount of red or excess steam used, the heating system should be inspected to learn of possible leakage. A large amount of blue graph means economical operation, and, as already suggested, it may be desirable to change the consumption unit for the following months to a slightly smaller unit consumption. **By means of careful observation the lowest possible unit steam consumption can be obtained, at the same time providing proper temperature in the building.**

The chart also has spaces for recording the day of the week, the time steam is turned on and off and a monthly summary applicable to either steam or fuel consumption. While much of the same data are recorded on the daily heating record (Fig. 1), it is considered desirable to have the chart show all of the essential facts for a month.

Fig. 3 shows a typical steam or fuel consumption chart for a building with a volume of approximately 10,000,000 cubic feet, the figures being based on steam consumption. This illustrates the use of the chart in plotting steam consumption from day to day.

The blank charts reproduced in Figs. 1 and 2 used by those who adopt this system of records are 11 x 17 inches in size.

This information was furnished by G. M. Tobey, who is secretary of the *Building Managers' Association of Chicago.*

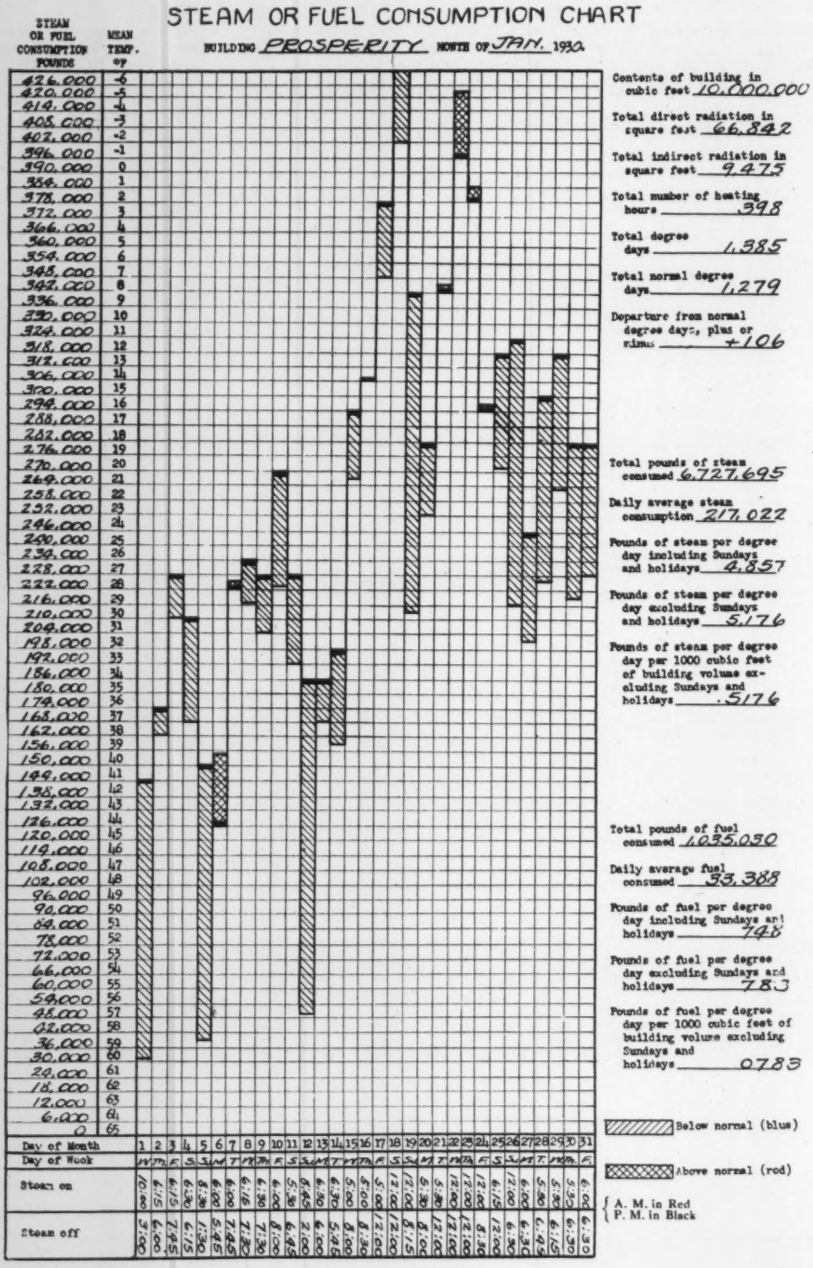
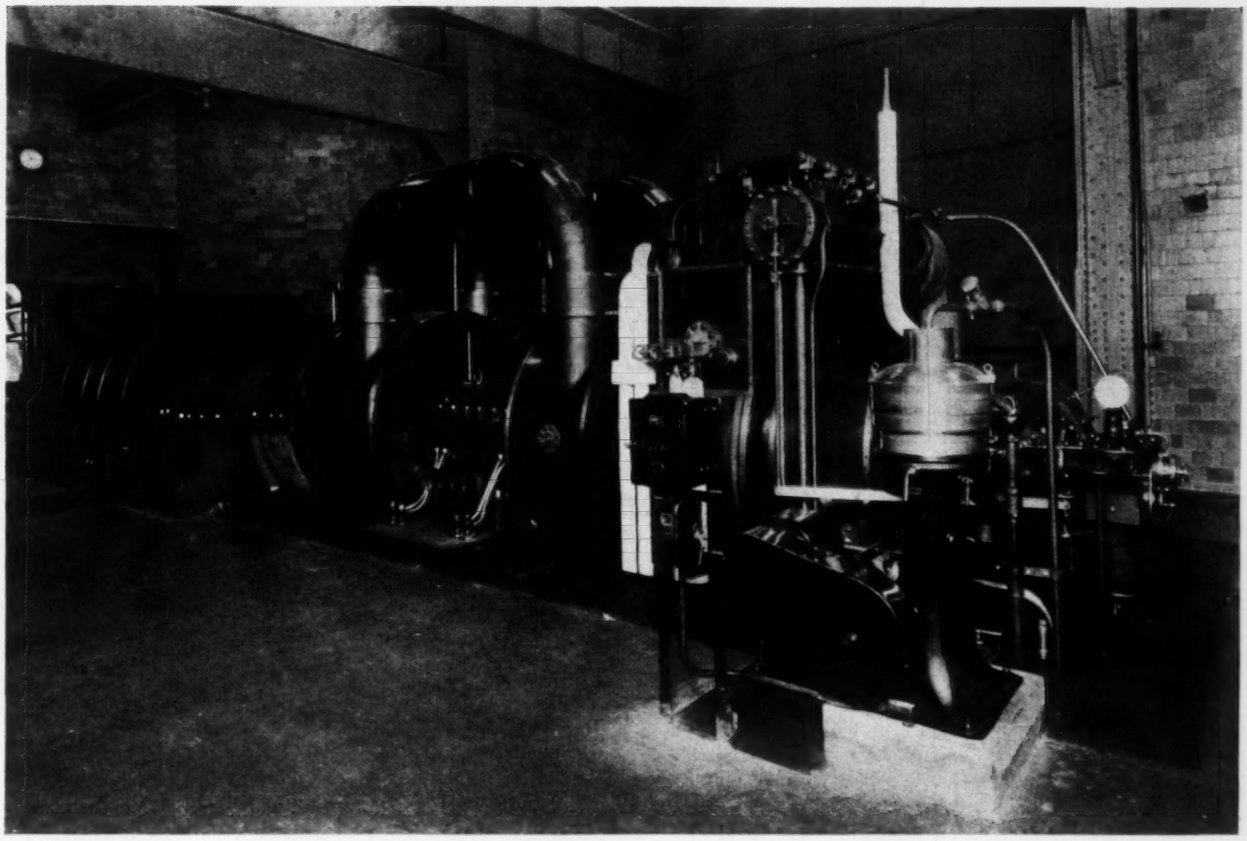

STEAM OR FUEL CONSUMPTION CHART
BUILDING PROSPERITY MONTH OF JAN. 1930
STEAM OR FUEL CONSUMPTION POUNDS
MEAN TEMP. °F
Contents of building in cubic feet 10,000,000
Total direct radiation in square feet 66,842
Total indirect radiation in square feet 9,475
Total number of heating hours 398
Total degree days 1,385
Total normal degree days 1,279
Departure from normal degree days, plus or minus +106
Total pounds of steam consumed 6,727,695
Daily average steam consumption 217,022
Pounds of steam per degree day including Sundays and holidays 4,857
Pounds of steam per degree day excluding Sundays and holidays 5,176
Pounds of steam per degree day per 1000 cubic feet of building volume excluding Sundays and holidays .5176
Total pounds of fuel consumed 1,035,030
Daily average fuel consumed 33,388
Pounds of fuel per degree day including Sundays and holidays 748
Pounds of fuel per degree day excluding Sundays and holidays 7.83
Pounds of fuel per degree day per 1000 cubic feet of building volume excluding Sundays and holidays 0783
Below normal (blue)
Above normal (red)
A. M. in Red
P. M. in Black
Day of Month
Day of Week
Steam on
Steam off

FIG. 3—TYPICAL STEAM OR FUEL CONSUMPTION CHART PLOTTED FOR A BUILDING WITH A VOLUME OF APPROXIMATELY 10,000,000 CUBIC FEET



10,000 KW. High Temperature Turbo-Generator

Designing High Temperature Steam Piping

Part I—Flexibility of 1000 F Line

By Arthur McCutchan*

This article tells how the piping for the turbo-generator at Delray No. 3 was designed; this unit operates on steam at 1,000 degrees Fahrenheit. ¶ This paper —in conjunction with others previously published in HEATING, PIPING AND AIR CONDITIONING and referred to in this article—explains the method used by The Detroit Edison Company in designing high temperature power plant piping

THE installation of a life-size turbo-generator unit designed to operate with steam at 1,000 F in the Delray power house of The Detroit Edison Company tests theories of design quite as severely as it does the metals of which the unit is constructed. A general description of the piping and the problems encountered in its design was given in an article entitled "Piping for 1,000-Degree Steam," by J. H. Walker, published in the April, 1931, Heating, Piping and Air Conditioning.

Engineering studies involved in the actual lay-out of this piping to provide for movement due to expansion and a means of compensating for the effects of "creep" on certain of the flanged joints have more than a nominal interest. The problems are not new but their importance is greatly emphasized by the increase of approximately 300 degrees in the normal operating temperature of the materials.

Conditions in a central station power plant are such

* Engineer, Engineering Division of The Detroit Edison Company, Detroit, Mich.

that equipment to be satisfactory must operate continuously for long periods. This requirement of continuity of service makes it difficult if not impossible to apply directly principles and practices which may be more or less established in other industries. Thus a type of flanged joint suitable for oil refinery service, for example, may prove wholly inadequate when applied to a main steam line in a power plant.

The general lay-out of the high temperature superheater, piping and turbo-generator is shown in Fig. 1, which is an isometric sketch of the installation.

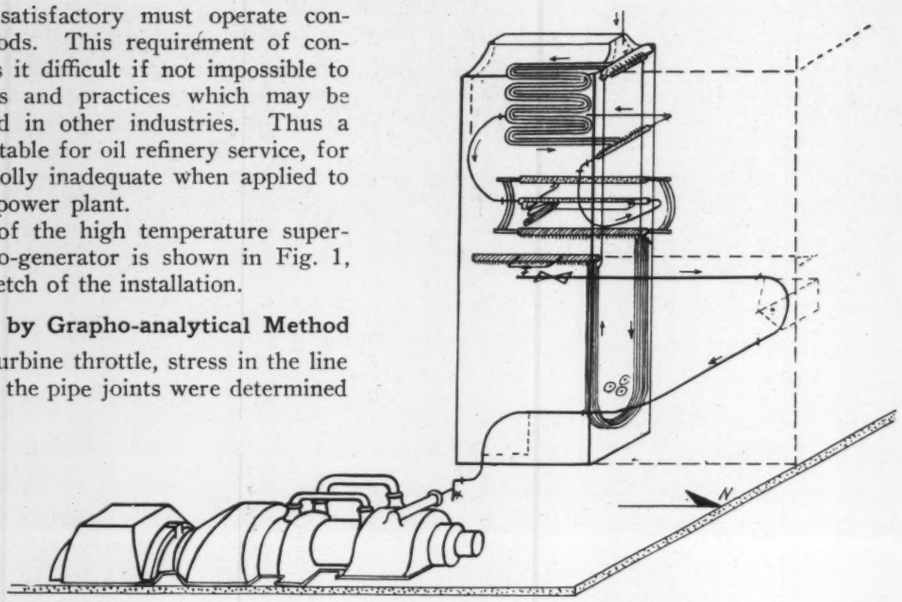


FIG. 1 — DIAGRAM OF THE 10,000 KW. HIGH TEMPERATURE UNIT INSTALLED AT DELRAY

Reactions Determined by Grapho-analytical Method

The reactions at the turbine throttle, stress in the line and bending moments at the pipe joints were determined by the grapho-analytical method described in some detail in an article on "The Design of Piping to Secure Flexibility" by the present writer in the September, 1930, HEATING, PIPING AND AIR CONDITIONING. In that article the calculation of the flexibility of an unsymmetrical pipe line lying in one plane was carried through as an illustration of the application of this method.

In order to provide a working model for the calculation of a pipe line in space, it appears desirable to publish the details of the calculations involved in the practical problem of analyzing the flexural characteristics of this 1,000 F steam line. This is especially desirable in view of the many statements current in piping literature to the effect that only a very rough approximation is possible for a pipe line which has runs in more than one plane. For example, in the description of the piping lay-out for the new Issy Les Moulineaux Station for the Paris Electric Co., which is designed for throttle steam conditions of 626 lb. per sq. in. and 842 F, the statement is made that due to the impossibility of calculating the flexibility of piping in more than one plane it was necessary to provide a physical separation between the piping in a vertical plane and that lying in a horizontal plane. The rather ingenious means provided to accomplish this result, as well as to introduce additional flexibility, is shown in Fig. 2. While the use of a bundle of small tubes in this manner has certain advantages when hard pressed to obtain adequate flexibility in a given line, as a general principle it is doubtful if such an expedient would be well received in this country.

Piping Has High Coefficient of Expansion

At Delray the high temperature line from the superheater to the turbine consists of approximately 43 linear feet of 8-in. pipe (8.625 in. O.D., 0.5 in. wall thickness) arranged as shown in Fig. 1. The material is a very low carbon variation of the alloy popularly known as 18-8 (steel containing 18 per cent chromium and 8 per cent nickel). This high-chromium, high-nickel austenitic alloy has demonstrated unusual load carrying ability at high temperatures. The exact composition of the pipe material as well as the other materials used in

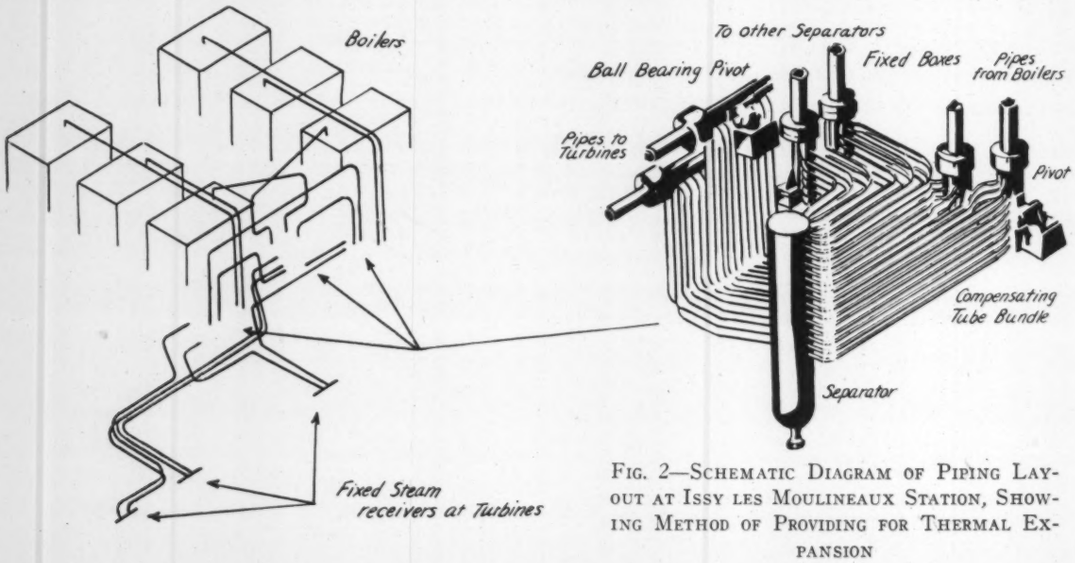


FIG. 2—SCHEMATIC DIAGRAM OF PIPING LAYOUT AT ISSY LES MOULINEAUX STATION, SHOWING METHOD OF PROVIDING FOR THERMAL EXPANSION

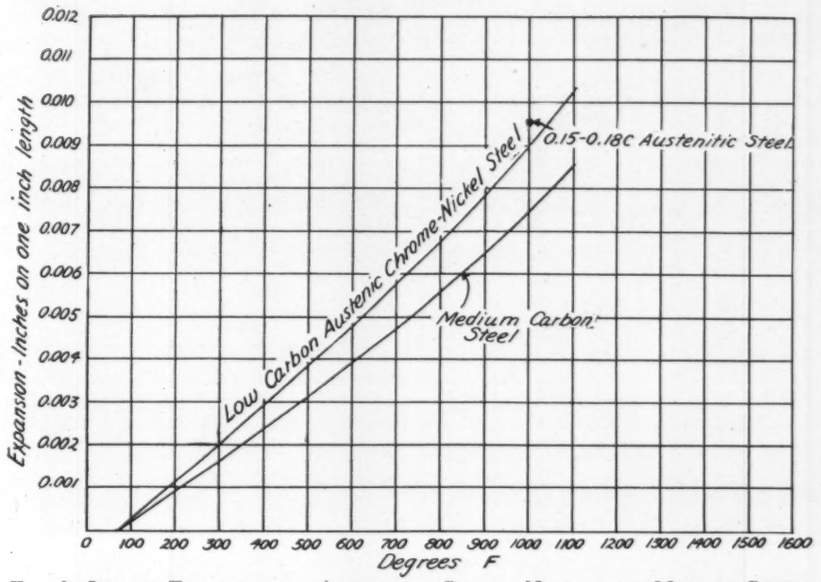


FIG. 3—LINEAR EXPANSION OF AUSTENITIC CHROME-NICKEL AND MEDIUM CARBON STEELS

this installation may be found in the tabulation given in the article by Mr. Walker.

A distinct characteristic of this type of alloy is an unusually high coefficient of linear expansion. The elongation which each inch of this material undergoes when heated from room temperature to a given operating temperature is plotted for comparison with carbon steel in Fig. 3. The writer believes this form of presenting expansion data to be preferable to that usually used which reports an average coefficient for a given temperature range, as the particular range seldom exactly meets the designer's requirements and some error naturally results.

When considering expansion stresses set up by temperature differences between flanges and bolting (as in *Part II* of this article) this form of expansion curve will be found particularly convenient. In the case of carbon steel pipe material there is considerable discussion at the present moment as to what linear expansion can be considered as representative of a given class of pipe material and the *Bureau of Standards* is co-operating with the *A. S. T. M.* toward clearing up discrepancies in published data. The values given in Fig. 3 are in agreement with the data given in the "Piping Handbook"[1] and are checked by values reported by manufacturers for very low carbon 18-8 material and for carbon steel. (See discussion of coefficients of linear expansion of piping in the October, 1929, HEATING, PIPING AND AIR CONDITIONING, p. 529.)

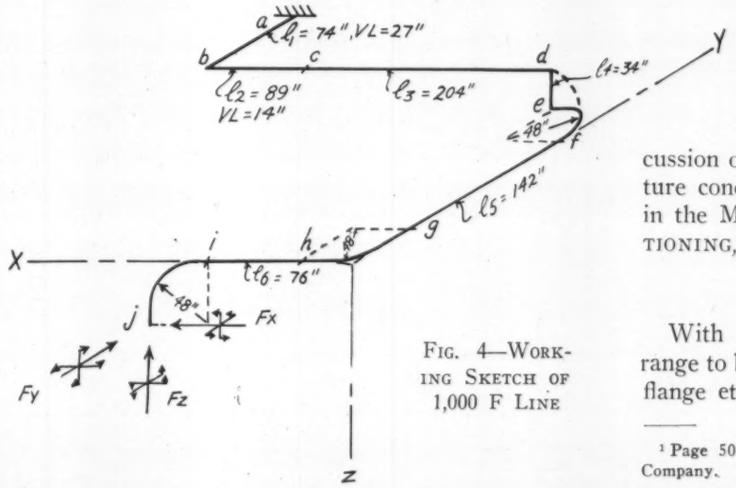


FIG. 4—WORKING SKETCH OF 1,000 F LINE

Design Conditions

The high temperature line is designed for operation at 1,000 F with provision for a possible ultimate operating temperature of 1,100 F. The ends of the line are considered as completely restrained from rotation at the superheater outlet header point *a*, and at the cast reducing elbow which is attached to the turbine stop and trip throttle valve flange, point *j* (See Fig. 4). The expansion to be taken care of is computed for the distances measured along the *X*, *Y* and *Z* axes between the center of the superheater outlet header and the 8-in. flange of the reducing elbow.

The expected movement of the throttle flange and elbow from the cold to the hot condition, as estimated by the turbine manufacturer, is taken into account in determining the expansion that must be provided for.

The allowable thrust on the throttle flange in any direction is given as 2,000 lb. by the turbine manufacturer. The total allowable working stress, considering bending plus longitudinal pressure stress was tentatively placed at from 6,000 to 7,000 lb. per sq. in. (See discussion of significance of bending stress under temperature conditions involving large "creep" of the material in the May, 1931, HEATING, PIPING AND AIR CONDITIONING, page 399).

Expansion to Be Cared For

With the foregoing assumptions as to temperature range to be considered, expected movement of the throttle flange etc., and the distances from Fig. 1, the elonga-

[1] Page 500, *Piping Handbook*, Walker and Crocker, McGraw-Hill Book Company.

tions from Fig. 3 are found to give the following potential deflections at 1,000 F.

$\Delta x = 0.39$ inches

$\Delta y = 3.18$ inches

$\Delta z = 1.67$ inches

Flexural Rigidity of Line at 1,000 F

The modulus of elasticity of pipe materials decreases with increase in temperature. In the case of carbon steels and usual alloy steels the value of the modulus at 1,000 or 1,100 F may be obtained with fair accuracy from such curves as those of Fig. 5. For the austenitic alloy steel used in this pipe line, less definite data are

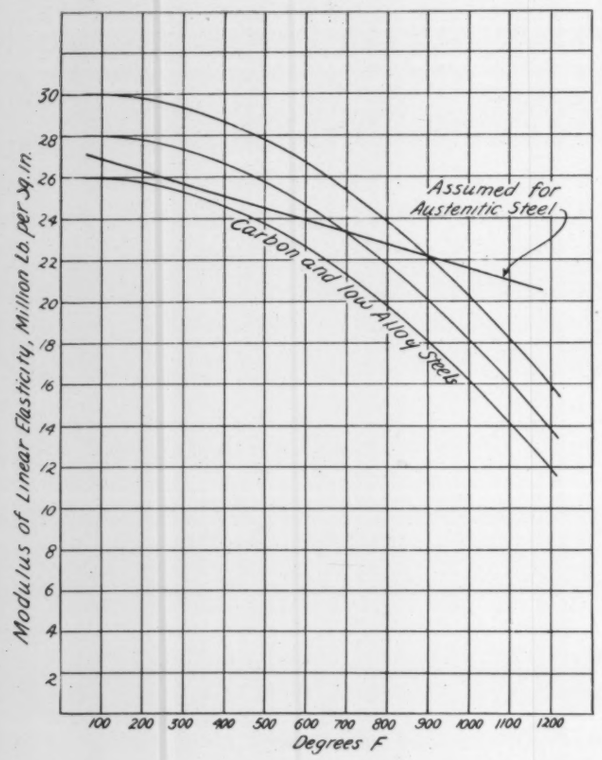


Fig. 5—Reduction in Modulus of Elasticity with Increase in Temperature

available at elevated temperatures. Hence, in order to be on the side of safety—i.e., not impute a degree of flexibility to the line which it might not possess—the straight line formula proposed by George Orrok [2] was used as a basis and the curve marked austenitic steels superimposed on the curves for carbon and low alloy steels. The modulus of elasticity determined on this basis is 21.6×10^6 lb. per sq. in. at 1,000 F.

The moment of inertia of the 8.625 O.D., ½-in. wall, pipe is 106.5 inches to the fourth power.

The flexural rigidity of the pipe in the straight parts of the line, which is the product of these two factors E and I, is $2{,}300 \times 10^6$.

The bends in the line are all formed to a radius of four feet which gives a ratio of mean radius of bend to pipe diameter of 6. This value corresponds to a K value of 0.746 as determined from Equation 5 on page 401 of the May, 1931, Heating, Piping and Air Conditioning given in the article on "Bending of Curved Pipes," by Sabin Crocker. The flexural rigidity of the curved sections of pipe with reference to bending in the plane in which the bend is formed is therefore given by $KEI = 1{,}720 \times 10^6$.

Accounting for Flexibility of Pipe, Valve, and Fittings Between Outlet Header and the Pipe Line Proper

The assembly of pipe, center pressure valve, and fittings between the outlet header of the superheater and the pipe line proper possesses a certain degree of flexibility which was taken into account as follows: The moments of inertia of the several parts were calculated and the actual lengths reduced in the ratio of their moments of inertia to the moment of inertia of the pipe. The resulting reduced lengths known as "virtual lengths"—or lengths of equivalent flexibility—are indicated on Fig. 4. One of the bends, *d-f*, actually is inclined at an angle to the horizontal plane, but for practical purposes its equivalent flexibility was considered as represented by a horizontal bend and a vertical length of pipe.

Grapho-analytical Method for One Force Described

The details of the grapho-analytical method of calculating the flexibility of the 1,000 F line can not be presented as completely as might be desired due to space limitations, but the procedure will be described for the effect of one force, the F_x force, acting in the XY plane. An examination of Fig. 4 shows that as far as bending of any part of the line by the F_x force is concerned in the XY plane, the first effect is noted at point h which tends to move to the left along the X-axis and in along the Y-axis. At point g the effect of the F_x force is represented by a force and a moment both of which tend to move point g to the left.

The bending moments due to the F_x force are shown on Fig. 6. Since the line is rigidly anchored or prevented from rotation at the ends, the algebraic sum of the bending moment areas which represent rotations of the component parts of the line must be zero in any plane. This fact enables the effect of the restraining moment to be determined in terms of the acting force by simply adding up the bending moment areas above the line and dividing by the sum of the lengths of the elements composing the base line.

Lengths Reduced in Proportion to Moments of Inertia

The length *a-b* is actually 74 inches, but as this portion of the line is composed of fittings and pipe much stiffer than the pipe its length is reduced in proportion to the moments of inertia of the fittings and pipe. The resulting length of 27 inches indicates the length of pipe which it is estimated would give the same deflection for a given force as would be produced on the greater length of more rigid pipe and fittings. In the same manner the length *b-c*, which includes the center pressure valve, is reduced from 89 inches to 14 inches. Length *c-d* is the actual length of pipe, while *d-e* which is assumed as 34 inches long is multiplied by a factor 1.3 which converts from torsional rigidity to rigidity in bending. *e-f* is the length of the pipe in the bend divided by

[2] *A. S. M. E. Trans.* (Fuels and Steam Power Section), May-Aug., 1928.

the rigidity multiplication factor K which corrects for the increase in flexibility of a curved pipe caused by flattening of the cross-section of the pipe when bending in the plane in which the bend is formed. The remaining lengths are found in the same manner, except the bend i-j which lies in the XZ plane. The restraining moment which prevents rotation of the end due to the F_X force acting in the XY plane is a torsional moment, hence its length is represented by 1.8 times the radius.

Resultant Bending Moments

After the bending moment diagram is completed for each force in each plane in which it has an effect, similar bending moment areas above and below the base line are cancelled leaving the resultant bending moment areas for each length. Each diagram represents the rotation of the elements of each part due to a force or moment acting at the end of the individual part. Thus, in the case of the length a-b, the triangular area represents a force acting on the end of a cantilever whose deflection is the product of the area and 2/3 of the actual length of a-b or 74 inches. The rectangular area represents the rotation of the part due to a moment acting at b. The deflection of b due to this moment is the product of the area and ½ the actual length. The total deflection of b due to the F_X force in the XY plane is the sum of the deflections due to the force and moment. In the same manner the deflection of the point c is obtained due to the bending moment acting at c. In addition to deflection of c due to bending within its length, there is an additional deflection to be added due to the fact that the end b is attached to the piece a-b which has been rotated due to the force and moment acting on it. The rotation of a-b is represented, as stated in the foregoing, by the triangular and rectangular bending moment areas. It is only necessary to add these areas together and multiply by the actual distance of c from b to find the deflection of c due to rotation of the elements of the part a-b. The total deflection of c is then the sum of the deflection due to bending within the length b-c and that found from the deflection due to rotation of the elements of a-b.

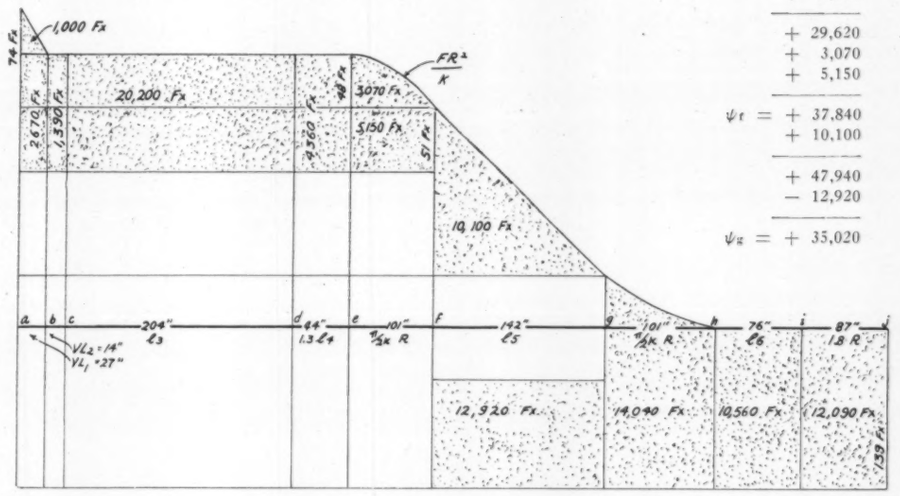


Fig. 6 — Bending Moment Diagram for F_x Force in XY Plane

Tabulation of Rotations

It is convenient to make a tabulation of the cumulative rotations for use in determining deflections of each part due to rotation of the part to which it is fastened. Since the sum of the rotations from one end of the line to the other must be zero (as explained in constructing the bending moment diagram) this tabulation gives an indication of the accuracy of the work. For Fig. 6 this tabulation is as follows:

Rotations Due to F_x Force in XY Plane

ψ_b =	+ 3,670	ψ_g =	+ 35,020
	+ 1,390		+ 1,750
ψ_c =	+ 5,060		+ 36,770
	+ 20,200		− 14,040
ψ_d =	+ 25,260	ψ_h =	+ 22,730
	+ 4,360		− 10,560
	+ 29,620	ψ_i =	+ 12,170
	+ 3,070		− 12,090
	+ 5,150		80
ψ_f =	+ 37,840		
	+ 10,100		
	+ 47,940		
	− 12,920		
ψ_g =	+ 35,020		

Tabulation of Deflections

The deflections of each point are then tabulated by identifying the force or moment acting on any point with similar forces and moments given in a series of elementary cases published in the "Piping Handbook."[3]

A tabulation of the deflections due to the F_X force in the XY plane is given below. The case numbers to which they correspond are indicated for reference

Deflections Due to F_x Force in XY Plane

$EI\ \Delta x_b$ =	Case II	$= \psi_F\ ⅔\ l_1$	$= 1{,}000 \times 49.3 = 49{,}300\ Fx$ left
	Case III	$= \psi_M\ ½\ l_1$	$= 2{,}670 \times 37 = 98{,}800\ Fx$ left
$EI\ \Delta y_c$ =	Case I	$= \psi_b\ l_2$	$= 3{,}670 \times 89 = 326{,}600\ Fx$ out
	Case III	$= \psi_M\ ½\ l_2$	$= 1{,}390 \times 44.5 = 61{,}900\ Fx$ out
$EI\ \Delta y_d$ =	Case I	$= \psi_c\ l_3$	$= 5{,}060 \times 204 = 1{,}032{,}000\ Fx$ out
	Case III	$= \psi_M\ ½\ l_3$	$= 20{,}200 \times 102 = 2{,}060{,}000\ Fx$ out
$EI\ \Delta x_f$ =	Case I	$= \psi_e\ R$	$= 29{,}620 \times 48 = 1{,}421{,}800\ Fx$ left
	Case Va	$= \psi_F\ 0.7854R$	$= 3{,}070 \times 37.7 = 116{,}000\ Fx$ left
	Case VIIa	$= \psi_M\ 0.637R$	$= 5{,}150 \times 30.6 = 157{,}500\ Fx$ left
$EI\ \Delta y_f$ =	Case I	$= \psi_e\ R$	$= 29{,}620 \times 48 = 1{,}421{,}800\ Fx$ out
	Case Vb	$= \psi_F\ 0.5R$	$= 3{,}070 \times 24 = 73{,}700\ Fx$ out
	Case VIIb	$= \psi_M\ 0.363R$	$= 5{,}150 \times 17.45 = 89{,}800\ Fx$ out
$EI\ \Delta x_g$ =	Case I	$= \psi_f\ l_5$	$= 37{,}840 \times 142 = 5{,}373{,}300\ Fx$ left
	Case II	$= \psi_F\ ⅔\ l_5$	$= 10{,}100 \times 94.5 = 955{,}000\ Fx$ left
	Case III	$= \psi_M\ ½\ l_5$	$= 12{,}920 \times 71 = 917{,}300\ Fx$ right
$EI\ \Delta x_h$ =	Case I	$= \psi_g\ R$	$= 35{,}020 \times 48 = 1{,}681{,}000\ Fx$ left
	Case VIa	$= \psi_F\ 0.624R$	$= 1{,}750 \times 29.9 = 52{,}400\ Fx$ left
	Case VIIb	$= \psi_M\ 0.363R$	$= 14{,}040 \times 17.4 = 244{,}600\ Fx$ right
$EI\ \Delta y_h$ =	Case I	$= \psi_g\ R$	$= 35{,}020 \times 48 = 1{,}681{,}000\ Fx$ in
	Case VIb	$= \psi_F\ 0.876\ R$	$= 1{,}750 \times 42.1 = 73{,}600\ Fx$ in
	Case VIIa	$= \psi_M\ 0.637R$	$= 14{,}040 \times 30.6 = 429{,}300\ Fx$ out
$EI\ \Delta y_i$ =	Case I	$= \psi_h\ l_6$	$= 22{,}730 \times 76 = 1{,}727{,}500\ Fx$ in
	Case III	$= \psi_M\ ½\ l_6$	$= 10{,}560 \times 38 = 401{,}300\ Fx$ out
$EI\ \Delta y_j$ =	Case I	$= \psi_i\ R$	$= 12{,}170 \times 48 = 584{,}200\ Fx$ in
	Case Xc	$= \psi_{M_t}\ 0.280R$	$= 12{,}090 \times 13.4 = 162{,}500\ Fx$ out

$EI\ \Delta x = 8{,}743{,}200$ Fx left

$EI\ \Delta y = 1{,}992{,}600$ Fx out

[3] For calculations involving piping in more than one plane, see revised cases VIII, IX and X in the second edition of the Piping Handbook. These cases are also available in the closure of a paper on "Frictional Resistance and Flexibility of Seamless Tube Fittings Used in Pipe Welding," by Sabin Crocker and Arthur McCutchan, *A. S. M. E. Trans.* (Fuels and Steam Power Section), Sept.-Dec., 1931.

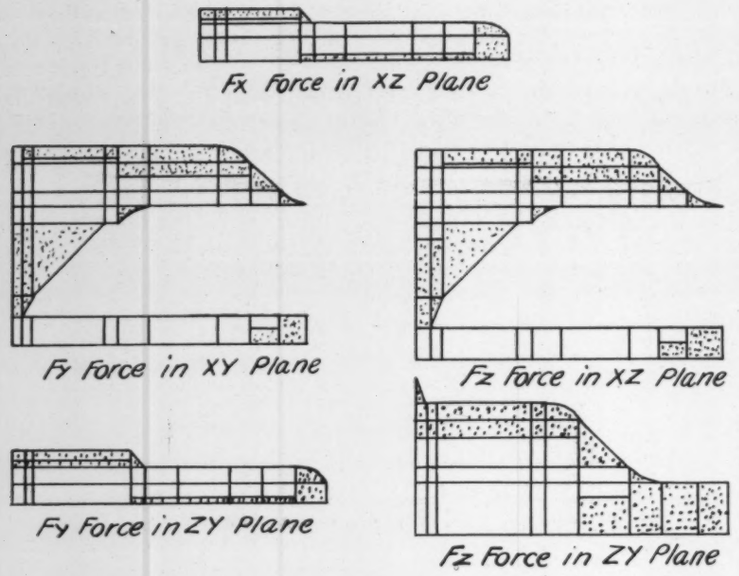


Fig. 7—Bending Moment Diagrams in Miniature

Similar tabulations must be made for each force in each of the two planes in which it causes movement. The bending moment diagrams for these remaining forces are shown in miniature in Fig. 7.

Numerical values for E, I and the expansions along the X, Y and Z axes, when substituted in the resulting equations form simultaneous equations from which the values of the reacting forces are found.

Reacting Forces in this Problem

In this problem the reacting forces along each axis are

$$F_X = 386 \text{ lb.}$$

$$F_Y = 1{,}210 \text{ lb.}$$

$$F_Z = 339 \text{ lb.}$$

Since the allowable thrust specified by the turbine manufacturer was 2,000 lb., the line as designed is satisfactory as far as reactions on the turbine throttle flange are concerned.

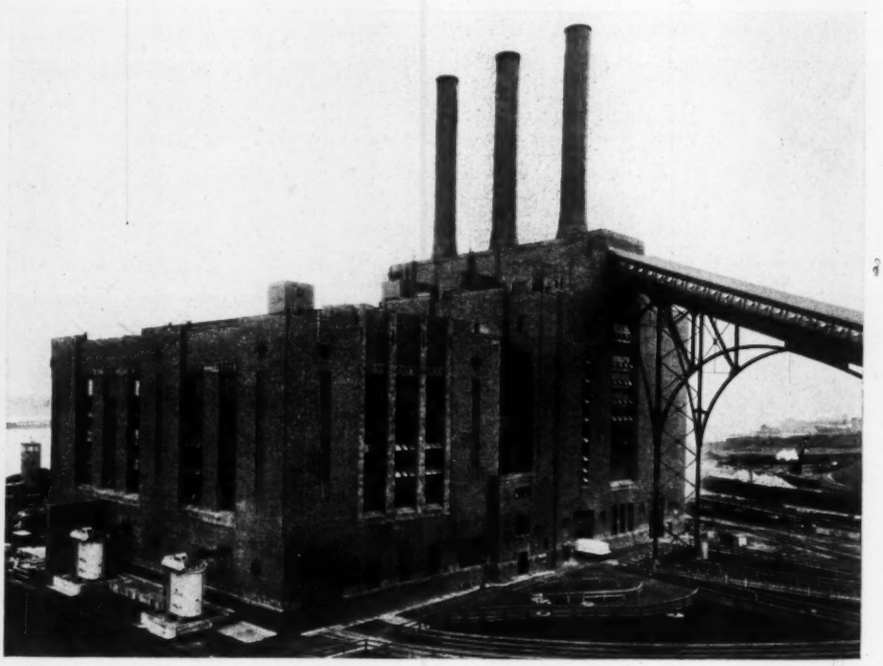
The values of the forces are substituted in the ordinates of the bending moment diagrams and the resulting bending moments determined at each point. In this case the maximum resultant bending moment in the 8-in. section of the line is found at point c. The bending stress at this point is calculated as 6,700 lb. per sq. inch. Actually the bending stress in the line will be somewhat less than calculated as the endeavor in all calculations of flexibility is to be assured that the stresses and reactions will not exceed certain values.

Assumption Errs on Safe Side

The assumption that the ends of the line are completely restrained from rotation, while sufficiently close to the actual condition, in most cases errs on the side of safety. The relative flexibility of an assemblage of valves and fittings is also a matter in which the judgment of the designer may influence the result. In the case of this high temperature installation the valves and fittings at either end of the line constitute a very considerable part of the line; also, the superheater outlet header is permitted by its supports to move along the X axis to a certain extent (Fig. 1). If an attempt is made to take into account all the stress reducing factors such as movement of the supports, flexure in the outlet header, etc., the resulting bending stress is reduced to approximately two-thirds that calculated on the foregoing assumptions. The location of the point of maximum bending stress is also changed from point c of Fig. 4 to point f by the different assumptions as to flexure of fittings and supports.

While in a particular installation it may prove desirable, as in this case, to attempt to evaluate and take advantage of the flexibility of involved fittings and valves, in general a sufficiently close approximation is possible without resorting to such analysis. A safe rule is to neglect the flexibility of parts of the equipment to which the line is joined which flexibility is at best somewhat problematical.

All that can be said with certainty of a result such as determined in the foregoing example is that the actual stresses and reactions will not exceed those determined.



First Section, Delray Power House No. 3

Shear Stress Due to Torsional Moment

In addition to the bending stress of 6,700 lb. per sq. inch, the shear stress due to a torsional moment acting at the same point would ordinarily be combined but as this stress amounts to only 104 lb. per sq. in. it may be neglected in this case. The longitudinal stress due to internal pressure amounts to 1,725 lb. per sq. in. which is directly additive to the bending stress. The resulting maximum stress in the line will not exceed 8,425 lb. per sq. in.

The calculation attempting to take into account the flexibility of the parts of the equipment acting in conjunction with the line indicates a total combined stress of only 6,240 lb. per sq. inch. The portion due to bending may be expected to relieve itself to a figure which the material is capable of sustaining. This reduction of bending stress due to the tendency of the line to accommodate itself to the shape of least resistance is an important consideration in designing piping for service at elevated temperatures.

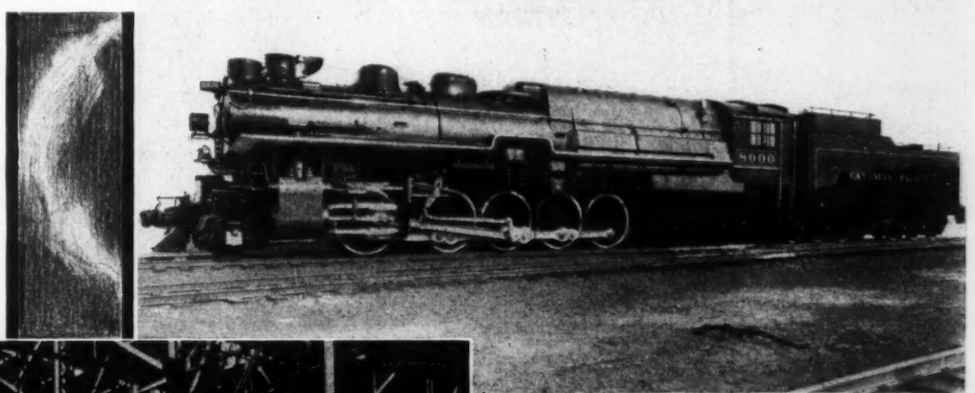
Since this reduction in stress with time is by its nature a lengthy process it is necessary to limit the initial stresses and reactions to values consistent with the known short time strength of the material at the operating temperature.

Editor's Note: Part II of this article will consider the design of flanged pipe joints for service at 1,000 F.

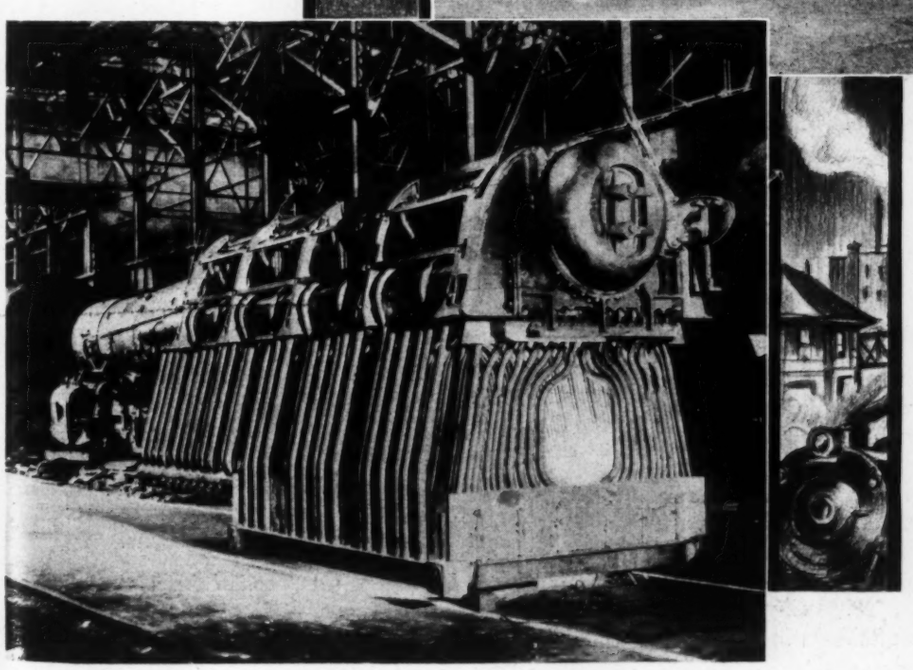
Canadian Pacific Locomotive Operates on 1350 Lb. Pressure

An example of the high pressures that are becoming more prevalent each year in steam power generators, in pipes and piping accessories, is illustrated here in what is said to be the largest and most powerful locomotive of its kind in the world.

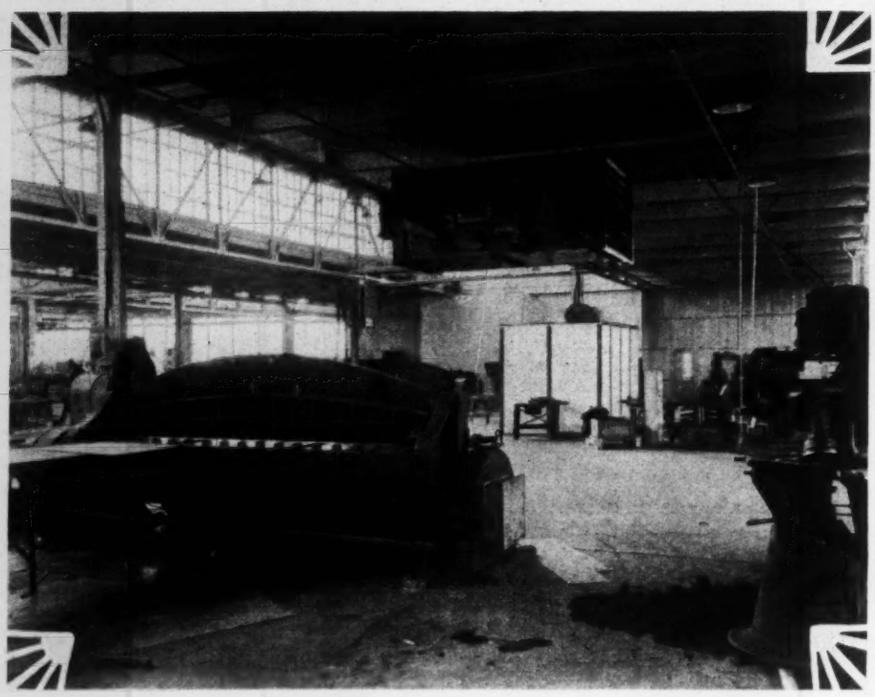
In its three and one-half miles of seamless steel tubing, 1,350 lb. per sq. in. pressure is maintained. It contains 300 gal. of distilled water in a closed circuit. The water is used over and over again and is not lost. The locomotive weighs 392½ tons and is 99 ft. long.



THE FIREBOX UNIT OF THE GIANT "8000" LOCOMOTIVE



NEW CANADIAN PACIFIC LOCOMOTIVE



7500 C.F.M. SUSPENDED TYPE UNIT AIR CONDITIONER USED IN A CIGAR FACTORY

By
Rush D. Touton*

Unit Conditioners

Supplement Central Air Conditioning Systems

MODERN industry, and particularly those branches of it which use raw materials of a hygroscopic nature, has come to accept air conditioning as an essential to satisfactory and economical production. The varied character of the services which must be considered under this one classification, including, as it does, the entire range of heating, cooling, humidifying and dehumidifying, is not so well appreciated.

The highly specialized cigar industry illustrates this variation well. For the efficient operation of a modern factory at least ten distinct atmospheric conditions must be maintained.

Wide Range of Conditions

The treatment of the general manufacturing areas, as an example, will require a dew point of approximately 68 F, while the proper condition for good fermentation will approach 110 F. Other processes, such as the storage of some types of prepared tobacco, are best handled at something like 50 F. Further than this, the unavoidable variation which is found in crops grown on the same plantations but under different weather conditions will make it necessary to change the processing conditions in accordance with the demands of the various tobaccos.

* Technical Director, Bayuk Cigars, Inc., Philadelphia, Pa.

Bayuk Cigars, Inc., uses about 25 unit air conditioners to supplement central systems.

These conditioners find application in industrial plants of all sizes; for the smaller manufacturing units, they furnish a satisfactory means of providing air conditioning where for economic reasons it is not possible with other equipment. The larger plants can utilize this equipment in areas and processes which, because of location, or the wide variation between the desired atmospheric conditions and those available from the central plant, are not practical to include in the main grouping of air conditioning systems.

A further advantage is found in the fact that these units are easily portable, and can usually be adapted to perform services of various kinds without great changes and expense. These features are of especial interest in the handling of seasonal work or the treatment of infrequent processes or shipments of special character.

Central Systems Used

Bayuk Cigars Inc. has systems in operation handling a total of well over 450,000 c.f.m. of air in its nine plants and warehouses. The greater part of this quantity is supplied by central station systems designed and installed for the particular purpose for which each is used. In its most recently erected plant, where 800,000 cigars are produced each day, the building was designed and constructed to provide the most favorable arrangement for thorough air conditioning and the systems were built in as an integral part of the structure. Sufficient refrigeration was provided to dehumidify the entire 2,035,000 cu. ft. of contents and to furnish the necessary cooling effect for a group of process rooms.

Use of Unit Conditioners

A number of situations have been found, however, where the central station equipment could not well be used. For these supplementary services unit humidifiers

offer a satisfactory solution. A wide variety of sizes and forms is available to meet practically all requirements.

One 5,000 c.f.m. unit is in operation without ducts and furnishes proper conditioning without drafts for all parts of a space 120 ft. long and 48 ft. wide. The fans provided are capable, however, of developing the necessary pressure for delivering the air through a system of ducts where it is required for proper diffusion.

Controls are available in several makes and forms, and the particular service for which the installation is made can be equipped with a completely automatic arrangement at a minimum of expense.

Bayuk Cigars Inc. have found advantageous applications for twenty-five units of 2,500 and 5,000 c.f.m. A description of their installations will, therefore, serve as an excellent illustration of the manifold uses of this equipment.

Fifteen Units Provide Necessary Flexibility for Curing Process

The erection of a completely humidified storage warehouse for the housing of the 7,000,000 pounds of tobacco required in their three-year natural curing process was a decided departure from former methods. The problem was handled by dividing the building into fifteen separate rooms, each one equipped with a 5,000-c.f.m. unit air conditioner. A pneumatic thermostat controls the admission of steam to the heating coils, which are of the extended surface type, and are installed as an integral part of the conditioner. An air-operated hygrostat controls the humidity by supplying steam to heat the spray water during the colder periods of the year, and in the summer when natural high humidities prevail this same instrument operates to turn the water supply to the humidifier on and off as the room conditions require.

The optimum of humidity and temperature is thus held during the long curing process, and the complete flexibility of the installation makes it possible to maintain fifteen separate atmospheric conditions for as many different lots of tobacco. This has proved to be very helpful in the proper development of the various tobaccos.

Maintaining Lower Dew Points Than in Main Part of Plants

Another instance was in providing humidification for the packing departments of two smaller plants. The desirable condition for this work is 64 F dew point, as compared to the 68 F carried on the cigar making floors of these plants. These departments, each with a floor space of approximately 6,000 sq. ft., are both equipped with 5,000-c.f.m. units. Both temperature and humidity are controlled by instruments operating suitable valves in the conditioner piping. The operation of these units has been entirely successful since their installation.

Conditioning for Department Which Is to Be Enlarged

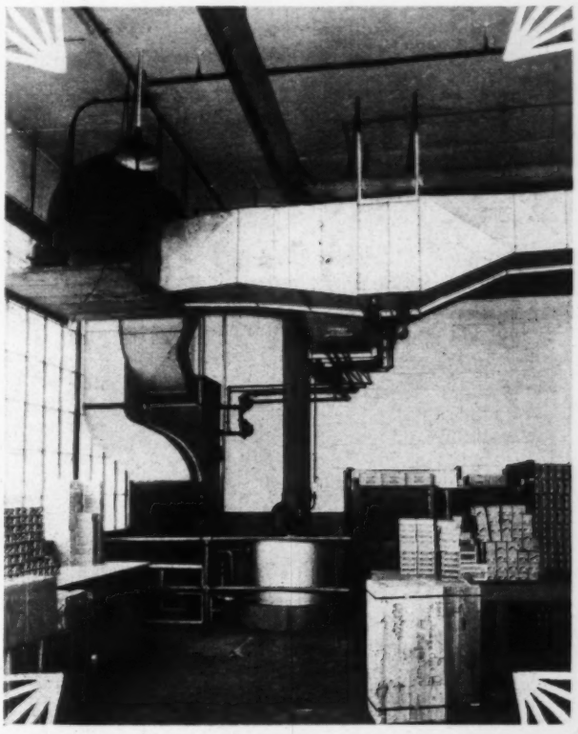
Another situation in which the installation of a central station humidifying apparatus was impossible has been adequately taken care of with a 2,500-c.f.m. unit. This was in a stripping department of a plant where an extension is to be made soon, with the result that any permanent equipment installed at this time would have to be completely altered and enlarged for the new service. The room has a floor space of 2,600 sq. ft., and forty workers are employed in it. There are several motors in operation in this area and forty 100-watt bulbs are used for lighting.

The same types of controls are used as in the large warehouse and provision was made for the use of either fresh or re-circulated air. The required 68 per cent relative humidity is easily maintained by the air delivered from the unit and the maximum heating load of 70,000 Btu per hour is well below the capacity of the heating surface in the conditioner.

Prepared Tobaccos Kept Cool and Moist

Some of the more costly types of tobaccos must be held in a moist condition for use in cigar making. In one of the Bayuk plants, a storage room of 10,000 cu. ft. was required for this purpose. The conditions in this room must be maintained at approximately 65 F and 85 per cent relative humidity. For this installation, a 5,000-c.f.m. unit was selected.

The walls of the storage room are built of 9-in. brick, and 2 in. of corkboard insulation was applied on the inner surface, with a ½-in. coat of portland cement plaster over it. A large number of openings were necessary to allow entry of the two conveyor lines, and, to avoid too great a waste of cooled and conditioned air, weighted flaps were placed in all apertures. The heat generated by the slight fermentation of the tobaccos during this storage period was also a factor in the calculations. Water is used as the cooling medium, supplied through insulated lines from the refrigeration equipment which is located in the basement, two floors below this level. An ejector duct is used to distribute the air in this



INDIVIDUAL AIR CONDITIONING EQUIPMENT FOR RETAIL DEPARTMENT

room. In this duct is placed a second fan, arranged so that it may recirculate an amount of the room air. The quantity of this recirculated air is regulated by a hygrostat, thus maintaining the relative humidity in the room at the desired point. A standard air-operated thermostat controls the amount of refrigerated water admitted to the suction of the spray pump and so controls the room temperature.

Unit Used in Conjunction With Larger Circulation System

The wholesale jobbing department of one of the Bayuk branches presented an unusual combination of air conditioning requirements. The space occupied is divided into two parts, one used for the office force of approximately twenty people, and the remainder of the space being used for storage of a comparatively large stock of cigars, and as a shipping room.

There existed a distinct need for humidification to insure proper care of the cigar stock, and to provide proper working conditions for the office force. For the same reasons, dehumidification was desirable for the summer months. A 2,500-c.f.m. unit was installed, with provisions for using fresh or return air or any mixture of the two. The heaters in the unit were not connected, but this air is delivered to the intake of a fan handling this 2,500 c.f.m. of air and an equal amount of recirculated air. This larger quantity of air is then delivered through separate duct systems to the two departments. Each one of the ducts is provided with an adequate amount of extended surface radiation to take care of the heating loads. Brine is used as the cooling medium, being piped from the refrigeration system located some distance away to a set of coils placed in an outer cooling tank adjacent to the conditioner. This tank is equipped with a separate agitator, which also elevates the cooled water to the suction of the spray pump. The cooling effect of about four tons of refrigeration is used in this service.

The adjustment of the fresh and return air mixing chamber is under manual control, and the correct humidity is obtained by a hygrostat operating a diaphragm valve in the steam line which heats the spray water. Flow of the cold brine into the coils of the water cooling tank is regulated by a self contained diaphragm valve with its thermostatic bulb placed in the water line to the humidifier.

Pre-Conditioning Arrangement for Correcting Regain of Tobaccos

From time to time it is found necessary to precondition various shipments of tobacco before opening the bales or cases in which the material is shipped. For this service, a separate room in one of the plants has been equipped with a 2,500-c.f.m. unit, and the proper connections have been made for supplying and controlling steam for both heating the spray water and the supply to the heating surface. Cold water is available for dehumidification.

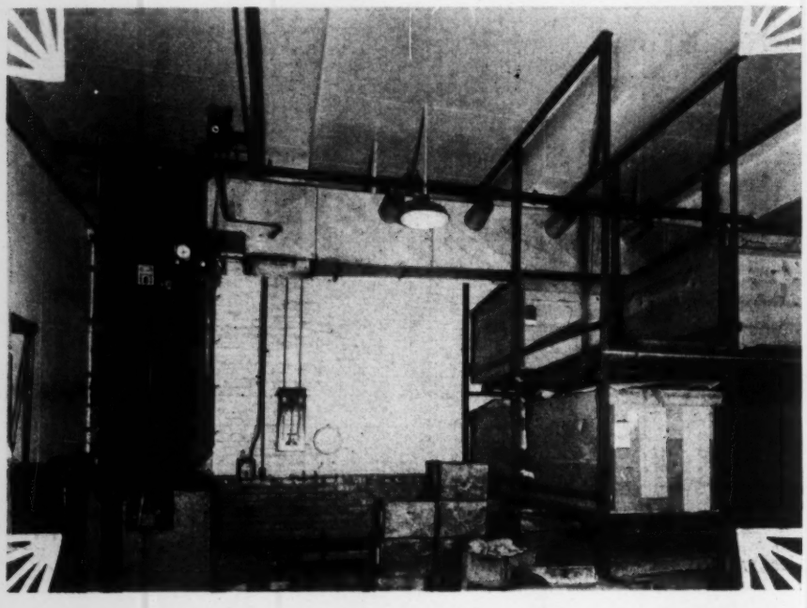
Into this room are taken bales, cases, bunches and separate leaves of tobacco. With the proper adjustment of the controls, any lot of the raw material which requires change in its moisture content is brought to the proper state for use. Since tobacco when too dry is extremely brittle and subject to breakage, and tobacco when too moist is inclined to ferment, the value of this room in normalizing these various moisture contents has been clearly evident.

Unit Conditioner Permits Additional Production

By installing a new and compact type of cigar machine in one of the Bayuk plants, sufficient floor space was made available for the production of 15 per cent more cigars in the same department. This was naturally the desirable thing to do. However, the air conditioning system, delivering 10,200 c.f.m. to this area, was already operating at its maximum capacity. Attempting to compensate for the increased load by raising the dew point of the air delivered by this equipment would obviously result in a room temperature too high for the comfort of the operators.

A 2,500-c.f.m. unit was installed near the point of greatest concentration of load. This occupies 9 sq. ft. of floor space, and is equipped with a 1 h.p. motor. The diffuser outlets on the original duct system were adjusted to deliver less air to this region, and so the two sources of supply work in co-ordination. A control on the new conditioner maintains the air delivered by the unit at a fixed dew point, and the hygrostatic control operating the larger original supply serves to take care of all fluctuations.

This addition of approximately 25 per cent to the total quantity of conditioned air, with a considerably lower increase in load, provides proportionately greater evaporative cooling effect and has also made it possible to reduce the dew point formerly carried.



A 5000 c.f.m. Unit Air Conditioner Showing By-Pass Arrangement for Control of Humidity

By changing some of the piping in connection with the refrigerating equipment, one manufacturing plant saved several hundred dollars a year in power charges for pumping. The computations leading to the decision to remodel the piping are presented here in detail, in order that they may be of assistance to other plants wishing to make similar savings.

Piping Improvements

in Condenser Water Circulating System

Reduce Cost of Pumping

By Herman Vetter*

WHEN machine parts are causing excess friction it is quickly evident. It is not so evident, however, when excess friction is created in water circulating lines; therefore, more alertness is required to detect these losses.

In a manufacturing plant which was recently remodeled numerous problems were met, one of which was the condenser water circulating system. An unusual number of sharp bends and other restrictions were noted in a preliminary inspection. The first test showed that the pump, which was rated at 750 g.p.m., was pumping only 500 g.p.m. against a head of 85 ft. and required 22.5 input horsepower, showing an overall efficiency of 50 per cent.

$$\frac{500 \text{ g.p.m.} \times 8.33 \text{ lb.} \times 89 \text{ ft.}}{33{,}000 \times 22.5 \text{ h.p.}} = 50\%$$

The pipe line friction head was found by deduction to be excessive:

Total Head		89.00
Static Head	29.00	
Spray Nozzles	16.10	
Condensers	3.04	
		48.14
Pipe Line Friction Head		= 40.86 ft.

In order to determine what changes should be made the pipe line was divided into the following sections and the friction head for each section calculated:

No. 1. Pump suction.
No. 2. Discharge between pump and condenser headers.
No. 3. Condenser headers.
No. 4. Condensers.
No. 5. Discharge from condenser to cooling tower.
No. 6. Spray header.

In calculating the friction head the following factors must be considered:

H, The velocity head.
H_1, Loss of head at entrance.
H_2, Loss of head through friction.
H_3, Loss of head due to obstructions.
H_4, Loss of head due to bends.

For comparison the difference in the calculated friction head between the original pump suction as shown in Fig. 2 and the new pump suction as shown in Fig. 1 will be shown.

* Consulting Engineer, Los Angeles, Calif.

Original Suction Line

Velocity Head

The velocity head is determined by the formula $H = v^2/2g$ where v = velocity in ft. per sec. and g = acceleration of gravity = 32.16 ft. per sec. per sec.

v = 9.8 ft. per sec. (600 g.p.m. through 5-in. pipe)

Therefore:

$$H = \frac{9.8^2}{2 \times 32.16} = 1.49 \text{ ft. head}$$

The velocity head is therefore 1.49 ft. for every 100 ft of pipe. The total length of the pump suction is 25 ft.; therefore, the velocity head is $1.49 \times 0.25 = 0.3725$ ft.

Entrance Loss

The loss of head at entrance is determined by the formula: $H_1 = K_1 \ v^2/2g$.

The values for K_1 are:

For inward projecting entrance,	0.78
For sharp edged entrance,	0.50
For slightly rounded entrance,	0.23
For bell-mouth entrance,	0.04

$$\therefore H_1 = 0.50 \frac{9.8^2}{2 \times 32.16} = 0.74 \text{ ft.}$$

Loss of Head Through Friction

The loss of head through friction is determined by the formula $H_2 = K_2 \ l/d \times v^2/2g$.

The value for K_2 for 5-in. pipe with water at 9.8 ft. velocity is 0.0222.

l = 25 ft. (length of pipe)

d = 0.416 ft. (inside dia. of pipe)

$$H_2 = 0.0222 \times \frac{25}{0.416} \times \frac{9.8^2}{2 \times 32.16} = 1.98 \text{ ft.}$$

Loss of Head from Obstructions

The loss of head due to obstructions is determined by the formula: $H_3 = K_3 \ v^2/2g$.

The obstructions existing in the line were due mostly to joints, and in calculating the loss of head it was assumed that the area of the pipe at the weld was reduced ⅛-in. in diameter.

K_3 = 0.10

$$H_3 = 0.10 \times \frac{9.8^2}{2 \times 32.16} = 0.1490 \text{ ft.}$$

Loss Due to Bends

The loss of head due to bends is determined by the the formula: $H_4 = K_4 \ v^2/2g$.

The value for K_4 for a standard 5-in. ell with water at 9.8 ft. per sec. velocity is 0.46.

$$H_4 = 0.46 \times \frac{9.8^2}{2 \times 32.16} = 0.6854 \text{ ft. per ell}$$

Two ells = $2 \times 0.6854 = 1.3708$ ft.

Total Friction Head

The total friction head for the pump suction was:

Velocity Head	H = 0.3725
Loss of Head at Entrance	H_1 = 0.7400
Loss of Head Through Friction	H_2 = 1.9800
Loss of Head Due to Obstructions	H_3 = 0.1490
Loss of Head Due to Bends	H_4 = 1.3708
	4.6123 ft.

New Suction Line

The pump suction was changed, as shown in Fig. 1. An 8-in. line with bell-mouth entrance was installed and a full size connection was made between this line and the pump. The suction connection on the pump was 6-in.

The total friction head on the new connection will be

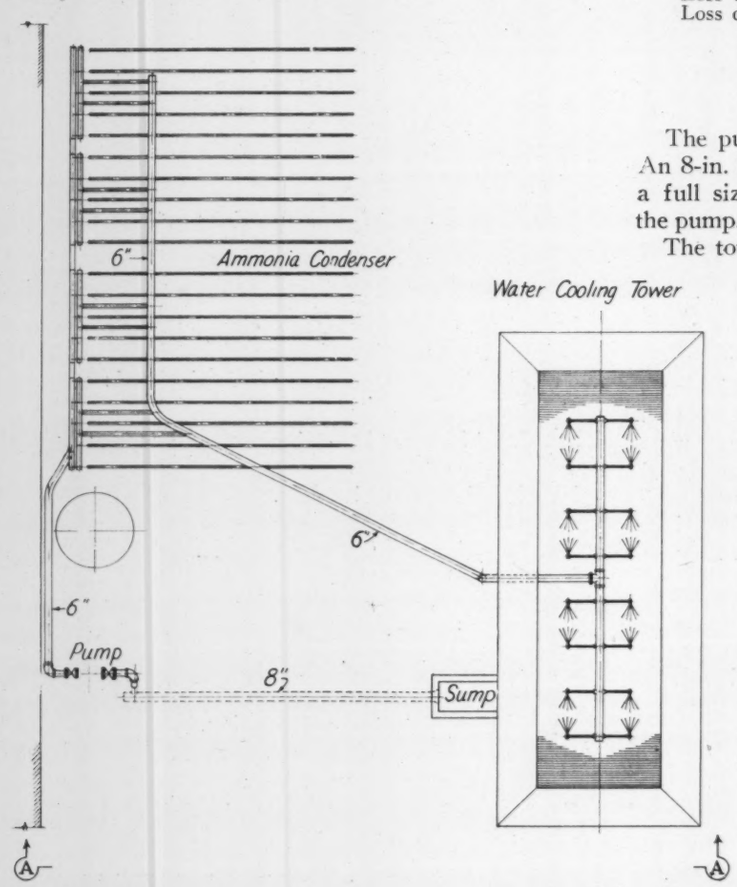


FIG. 1 — THE PUMP SUCTION WAS CHANGED, AS SHOWN IN THE DIAGRAM. AN 8-IN. LINE WITH BELL-MOUTH ENTRANCE WAS INSTALLED

calculated so as to compare it with the original suction connection.

Velocity Head (8-in. Pipe)

$v = 3.84$ ft. per sec. for 600 g.p.m. through an 8-in. pipe

$$H = \frac{3.84^2}{2 \times 32.16} = 0.2292 \text{ ft. head/100 ft. pipe}$$

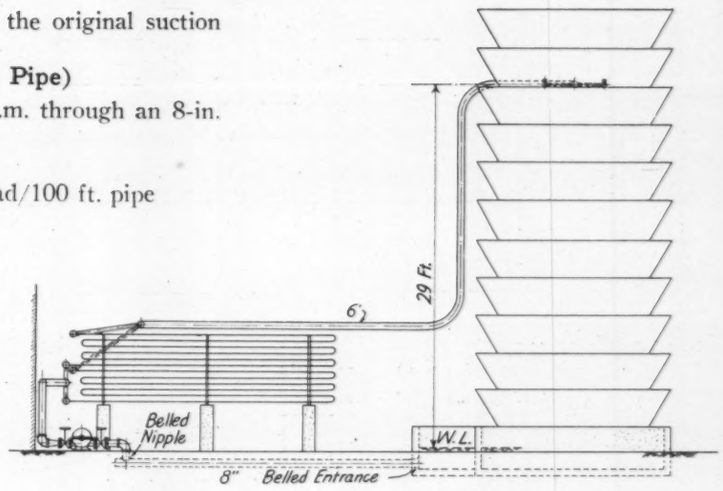


FIG. 1A — SECTION A - A THROUGH FIG. 1, SHOWING THE NEW PUMP SUCTION LINE

$0.2292 \times 0.23 = 0.0527$ ft. head for 23 ft. of 8-in. pipe.

Loss of Head at Entrance (Sump to 8-In. Pipe)

$$H_1 = 0.04 \times \frac{3.84^2}{2 \times 32.16} = 0.0092 \text{ ft.}$$

Loss of Head at Entrance (8-In. Pipe to 6-In. Pump Suction Conn.)

$v = 6.72$ ft. per sec. for 600 g.p.m. through a 6-in. pipe

$$H_1 = 0.23 \times \frac{6.72^2}{2 \times 32.16} = 0.1615 \text{ ft.}$$

Velocity Head (Pump Connection)

$$H = \frac{6.72^2}{2 \times 32.16} = 0.7020 \text{ ft. head/100 ft. pipe}$$

$0.7020 \times 0.05 = 0.0351$ ft. head for 5-ft. length of pipe.

Loss of Head Through Friction (8-In. Pipe)

$K_2 = 0.0234$ for 8-in. pipe at 3.84 ft. per sec. velocity
$l = 23$ ft.; $d = 0.666$

$$H_2 = 0.0234 \times \frac{23}{0.666} \times \frac{3.84^2}{2 \times 32.16} = 0.1952 \text{ ft. head.}$$

Loss of Head Through Friction (6-In. Pipe)

$K_2 = 0.0249$ for 6-in. pipe at 6.72 ft. per sec. velocity
$l = 5$ ft.; $d = 0.5$ ft.

$$H_2 = 0.0249 \times \frac{5}{0.5} \times \frac{6.72^2}{2 \times 32.16} = 0.1748 \text{ ft. head.}$$

Loss of Head Due to Obstructions (8-In. Pipe)

$K_3 = 0.10$

$$H_3 = 0.10 \times \frac{3.84^2}{2 \times 32.16} = 0.0229 \text{ ft.}$$

Loss of Head Due to Bends

$K_4 = 0.40$ for 6-in. ell at 6.72 ft. per sec. velocity

$$H_4 = 0.40 \times \frac{6.72^2}{2 \times 32.16} = 0.2808 \text{ ft. head for each ell.}$$

Total Friction Head

The total friction head in the new pump suction is:

The Velocity Head for 8-In. Pipe	$H = 0.0527$
The Velocity Head for 6-In. Pipe	$H = 0.0351$
Loss of Head at Entrance for 8-In. Pipe	$H_1 = 0.0092$
Loss of Head at Entrance for 6-In. Pipe	$H_1 = 0.1615$
Loss of Head Through Friction in 8-In. Pipe	$H_2 = 0.1952$
Loss of Head Through Friction in 6-In. Pipe	$H_2 = 0.1748$
Loss of Head Due to Obstructions	$H_3 = 0.0229$
Loss of Head Due to 6-In. Ell	$H_4 = 0.2808$
	0.9322 ft.

The reduction in the total friction head between the former pump suction and the new arrangement is $4.6123 - 0.9322 = 3.6801$ ft.

A similar comparison was made with the rest of the pipe line but as these calculations would be a repetition the totals only are given.

Total Friction Head for Discharge Between Pump and Condenser

Velocity Head for 25 Ft. 6-In. Pipe (600 g.p.m.)	$H = 0.1755$
Velocity Head for 8 Ft. 6-In. Pipe (450 g.p.m.)	$H = 0.0326$
Velocity Head for 8 Ft. 6-In. Pipe (300 g.p.m.)	$H = 0.0143$
Velocity Head for 8 Ft. 6-In. Pipe (150 g.p.m.)	$H = 0.0036$
Friction Head for 25 Ft. 6-In. Pipe (600 g.p.m.)	$H_2 = 0.8740$
Friction Head for 8 Ft. 6-In. Pipe (450 g.p.m.)	$H_2 = 0.1539$
Friction Head for 8 Ft. 6-In. Pipe (300 g.p.m.)	$H_2 = 0.0698$
Friction Head for 8 Ft. 6-In. Pipe (150 g.p.m.)	$H_2 = 0.0192$
Friction Head for 1—6-In. Ell (600 g.p.m.)	$H_4 = 0.2808$
Friction Head for 3—6-In. 45 deg. Bends (600 g.p.m.)	$H_4 = 0.3948$
Entrance Head for 3-In. Outlet (75 g.p.m.)	$H_1 = 0.0378$
Loss of Head Due to Obstruction 6-In. Pipe (600 g.p.m.)	$H_3 = 0.2808$
	2.3371 ft.

Friction Head in Condenser

The friction head in the condenser and headers was then calculated as follows:

The condenser consists of twenty coils each twelve pipes high. The water section is split in two so that the water flows through six pipes only. The quantity of water passing through each coil is therefore 15 g.p.m.

Condenser inlet header:

Velocity Head for 2 Ft. of 6-In. Pipe (30 g.p.m.)	$H = 0.000020$
Velocity Head for 2 Ft. of 6-In. Pipe (15 g.p.m.)	$H = 0.000004$
Friction Head for 2 Ft. of 6-In. Pipe (30 g.p.m.)	$H_2 = 0.000127$
Friction Head for 2 Ft. of 6-In. Pipe (15 g.p.m.)	$H_2 = 0.000000$
Entrance Head for 1½-In. Inlet (15 g.p.m.)	$H_1 = 0.043280$
	0.043431 ft.

As the outlet header is the same as the inlet header the total head for the two headers is $0.0434 \times 2 = 0.0868$ ft.

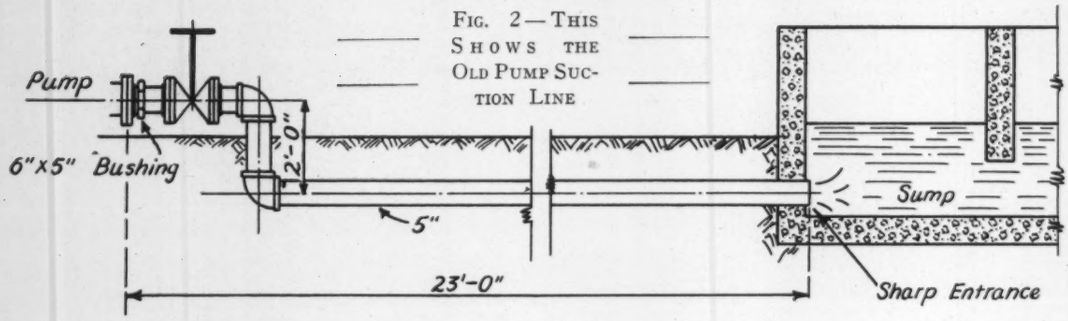


FIG. 2—THIS SHOWS THE OLD PUMP SUCTION LINE

Friction head in the condenser:

Velocity Head for 120 Ft. of 1¼-In. Pipe (15 g.p.m.)	= 0.1038
Friction Head for 120 Ft. of 1¼-In. Pipe (15 g.p.m.)	= 2.4248
Loss of Head for 10 Bends (90 deg.) (15 g.p.m.)	= 0.4410
	2.9696
Loss of Head in Headers	= 0.0868
Total Loss of Head	= 3.0564 ft.

Loss of Head from Condenser to Cooling Tower

The loss of head in the discharge from the condenser to the cooling tower is shown below:

Velocity Head for 6 Ft. of 3-In. Pipe (150 g.p.m.)	H = 0.0396
Friction Head for 6 Ft. of 3-In. Pipe (150 g.p.m.)	H_3 = 0.4118
Velocity Head for 8 Ft. of 6-In. Pipe (150 g.p.m.)	H = 0.0036
Friction Head for 8 Ft. of 6-In. Pipe (150 g.p.m.)	H_3 = 0.0192
Velocity Head for 8 Ft. of 6-In. Pipe (300 g.p.m.)	H = 0.0143
Friction Head for 8 Ft. of 6-In. Pipe (300 g.p.m.)	H_3 = 0.0698
Velocity Head for 8 Ft. of 6-In. Pipe (450 g.p.m.)	H = 0.0326
Friction Head for 8 Ft. of 6-In. Pipe (450 g.p.m.)	H_3 = 0.1539
Velocity Head for 48 Ft. of 6-In. Pipe (600 g.p.m.)	H = 0.3370
Friction Head for 48 Ft. of 6-In. Pipe (600 g.p.m.)	H_3 = 1.6752
Friction Head for 3—6-In. Bends (600 g.p.m.)	H_4 = 0.3948
Loss of Head Due to Obstructions—6-In. Pipe	H_3 = 0.4212
	3.5730 ft.

Friction Head in Spray Header

We will now calculate the friction head in the spray header which will conclude the calculations, with the exception of totalling the various items.

Velocity Head for 4 Ft. of 6-In. Pipe (75 g.p.m.)	H = 0.0004
Friction Head for 4 Ft. of 6-In. Pipe (75 g.p.m.)	H_3 = 0.0025
Velocity Head for 4 Ft. of 6-In. Pipe (150 g.p.m.)	H = 0.0017
Friction Head for 4 Ft. of 6-In. Pipe (150 g.p.m.)	H_3 = 0.0093
Velocity Head for 4 Ft. of 6-In. Pipe (225 g.p.m.)	H = 0.0039
Friction Head for 4 Ft. of 6-In. Pipe (225 g.p.m.)	H_3 = 0.0196
Velocity Head for 2 Ft. of 6-In. Pipe (300 g.p.m.)	H = 0.0036
Friction Head for 2 Ft. of 6-In. Pipe (300 g.p.m.)	H_3 = 0.0174
Entrance Head for 1¼-In. Pipe (38 g.p.m.)	H_1 = 0.2798
Loss of Head for 6-In., 90-deg. Bend (300 g.p.m.)	H_4 = 0.0646
	0.4028 ft.

The Total

Below is shown the total of the various friction heads to which is added the static head and the head on the spray nozzles.

Pump Suction	= 0.9322
Discharge Between Pump and Condenser Headers	= 2.3371
Condenser Headers	= 0.0868
Condensers	= 2.9696
Discharge from Condenser to Cooling Tower	= 3.5730
Spray Header	= 0.4028
Friction Head	= 10.3015
Static Head	= 29.0000
Spray Nozzles = 7 lb. × 2.3	= 16.1000
Total Head	= 55.4015 ft.

Test of New Piping

In these calculations no allowance was made for water scale in the pipes; therefore, we will assume an allowance of ten feet for this resistance which gives a total estimated head of 65 ft. for circulating 600 g.p.m. as compared with 89 ft. for circulating 500 g.p.m. for the old piping.

After the piping changes were made and before a new pump was installed a test was run to prove what effect the pipe changes had accomplished and also to give definite data to determine whether it was advisable to change the pump. This test gave these results:

Capacity	740 g.p.m.
Head	82 ft.
Input power	26.5 h.p.
Efficiency	58 per cent

These results showed an increase of 240 g.p.m. (or an increase of 48 per cent) in capacity at an increase of only 17.8 per cent in power.

A New Pump?

The refrigerating capacity of this plant requires only 600 gallons of water per minute and the old pump (even with the impeller cut down to reduce its capacity to 600 g.p.m.) could only be guaranteed by the manufacturer to give an efficiency of 70 per cent, whereas a pump more suitable to the estimated capacity and head could be guaranteed to operate at 78 per cent efficiency.

The difference in estimated power consumption between the old pump and a new pump is shown below:

Old Pump

$$\frac{600 \text{ g.p.m.} \times 8.33 \text{ lb.} \times 65 \text{ ft.}}{33{,}000 \times 0.70} = 14.08 \text{ h.p.}$$

14.08 h.p. × 0.746 × 24 hrs. × 30 days = 7,560 kw-hr./mo.

New Pump

$$\frac{600 \text{ g.p.m.} \times 8.33 \text{ lb.} \times 65 \text{ ft.}}{33{,}000 \times 0.78} = 12.61 \text{ h.p.}$$

12.61 h.p. × 0.746 × 24 hrs. × 30 days = 6,770 kw-hr./mo.

7560 — 6770 = 790 kw-hr. @ 0.0075 = $5.93/mo.

As this pump is in operation continuously for 8 months each year the annual saving with a new pump would be $5.93 × 8 = $47.44.

The difference in cost between a new pump and reconditioning the old one was $350.00; therefore it was considered advisable to install a new pump.

Test Run After New Pump Was Installed

After the new pump was installed a final test was made with the following results.

Capacity	660 g.p.m.
Head	63.2 ft.
Power	13.36 h.p.
Efficiency	80 per cent

The difference between the estimated head and the actual head was 2.8 feet, which resulted in an increase of ten per cent in capacity. This difference could be corrected by reducing the diameter of the impeller. This was not done, however, as the slight increase in capacity and power was not considered objectionable.

Comparison of Old and New System

As the former pumping system was delivering only 83

per cent of the desired capacity a direct comparison can be made only by basing the comparison on the same quantity of water handled. This comparison is as follows.

Former Pumping System

$$\frac{22.5 \text{ h.p.}}{500 \text{ g.p.m.}} = 0.045 \text{ h.p.} \times 600 \text{ g.p.m.} = 27 \text{ h.p.}$$

New Pumping System

$$\frac{13.36 \text{ h.p.}}{600 \text{ g.p.m.}} = 0.022 \text{ h.p.} \times 600 \text{ g.p.m.} = 13.2 \text{ h.p.}$$

The pipe changes therefore netted a reduction of $27 - 13.2 = 13.8$ h.p.

Reduction in Power Cost

The actual reduction in power cost between the former and the new arrangement is:

22.5 h.p. × 0.746 × 24 hr. × 30 days × 8 mo. = 96,682 kw-hr.

96,682 kw-hr. @ 0.0075 = $725.12.

13.36 h.p. × 0.746 × 24 hr. × 30 days × 8 mo. = 57,240 kw-hr.

57,240 kw-hr. @ 0.0075 = $429.30

The net yearly saving in power is $725.12 — $429.30 = $295.82.

The intention of this article is not to show how any certain pumping problem should be handled but to show the possibilities of reducing manufacturing costs by eliminating unnecessary losses. The formulas used in the calculations are published in King's Handbook of Hydraulics. (John Wiley & Sons, Inc., New York City.)

THE CENTRIFUGAL PUMPS FOR THE COOLING SYSTEM AT KDKA

Piping System for

Power Tube Cooling Water at Radio Station KDKA

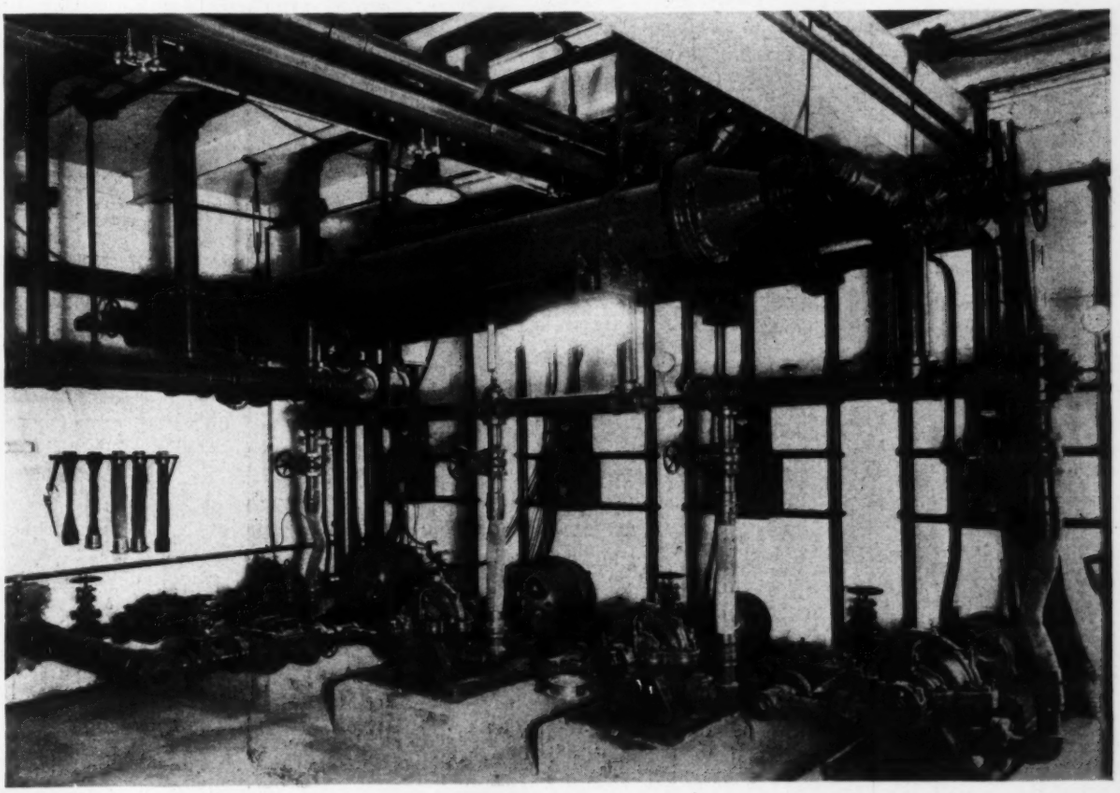
By E. M. Sollie*

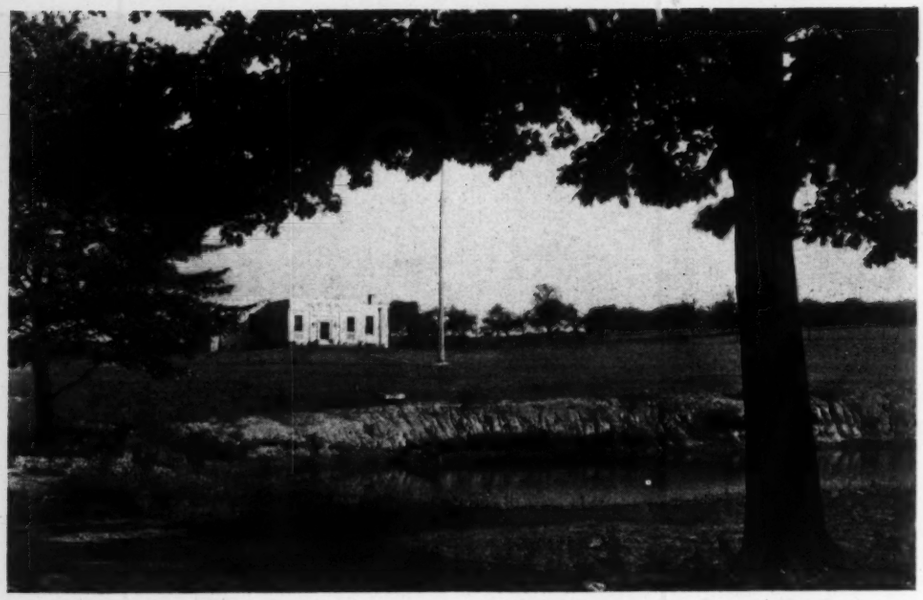
In the design of a transmitting station of the size of *KDKA* there were many problems to be solved in the selection of apparatus. The transmitter was built with the intention of making the installation available for experimental purposes on high power, as well as for regular broadcasts on 50 kw. One of the many varied problems arising in the design was that of the selection of a water cooling system for the power tubes.

A great deal of power must be dissipated by this system and great care was taken in the selection of the system. The filament power of each of the 100 kw. tubes is 10 kw., and for high power tests up to fourteen of these tubes are on at the same time, in addition to the smaller tubes in the intermediate amplifiers.

When operating as a 50 kw. transmitter, the water system must dissipate in the neighborhood of 125 kw.,

* Radio Operations Dept., Westinghouse Electric and Manufacturing Company, East Pittsburgh, Pa.





and when high power tests are conducted the water system must dissipate about 700 kw.

Two Water Systems

In the present layout of *KDKA* two water systems are used. One of these is used to furnish the cooling water for the power tubes and the second to cool the water flowing in the first system.

The tube water (as the first system is called) is stored in a sump located outside the station house. It is connected to the drains of the roof of the station house and thus utilizes rain water, which, because of its high resistivity, reduces electrolysis in the metal connections at the power tubes as much as possible. The water is obtained from this sump by means of either one or two centrifugal pumps which deliver the water to the dis-

At Top—Station KDKA at Saxonburg. The Pond Furnishing the Cooling Water Is Shown in the Foreground

Broadcast Power Amplifier Which Increases the Strength of the Signals and Sends Them on the Air. Notice the Four Large Amplifier Tubes. Mr. Sollie, Station Engineer, Makes Hourly Tests to Be Sure the Water Passing Through the Tubes Is Not Boiling. This Is Accomplished By Listening to the Flow of the Water Through a Hollow Micarta Tube

tribution system at a pressure of 65 lb. The water is fed to the tubes through a coil of hose of sufficient length to cut down the power loss and the electrolysis to a negligible amount. The water is removed from the tube in a similar way. After the water comes from the tubes, it flows by gravity through a heat interchanger back to the sump. This heat interchanger consists of a large number of small tubes mounted between two end plates (similar to steam boiler construction) and placed in a long iron pipe, or mounting, about 1½ ft. in diameter. The water from the tubes runs around these small pipes, and then flows back to the sump.

In order to obtain cooling water to be used in this heat interchanger, a large pond was constructed in front of the station house, approximately 150 ft. long, 75 ft. wide, and 8 ft. deep. It is supplied with water by seepage and surface drainage. One of the determining factors in the station's location was the ample supply of water available at all times.

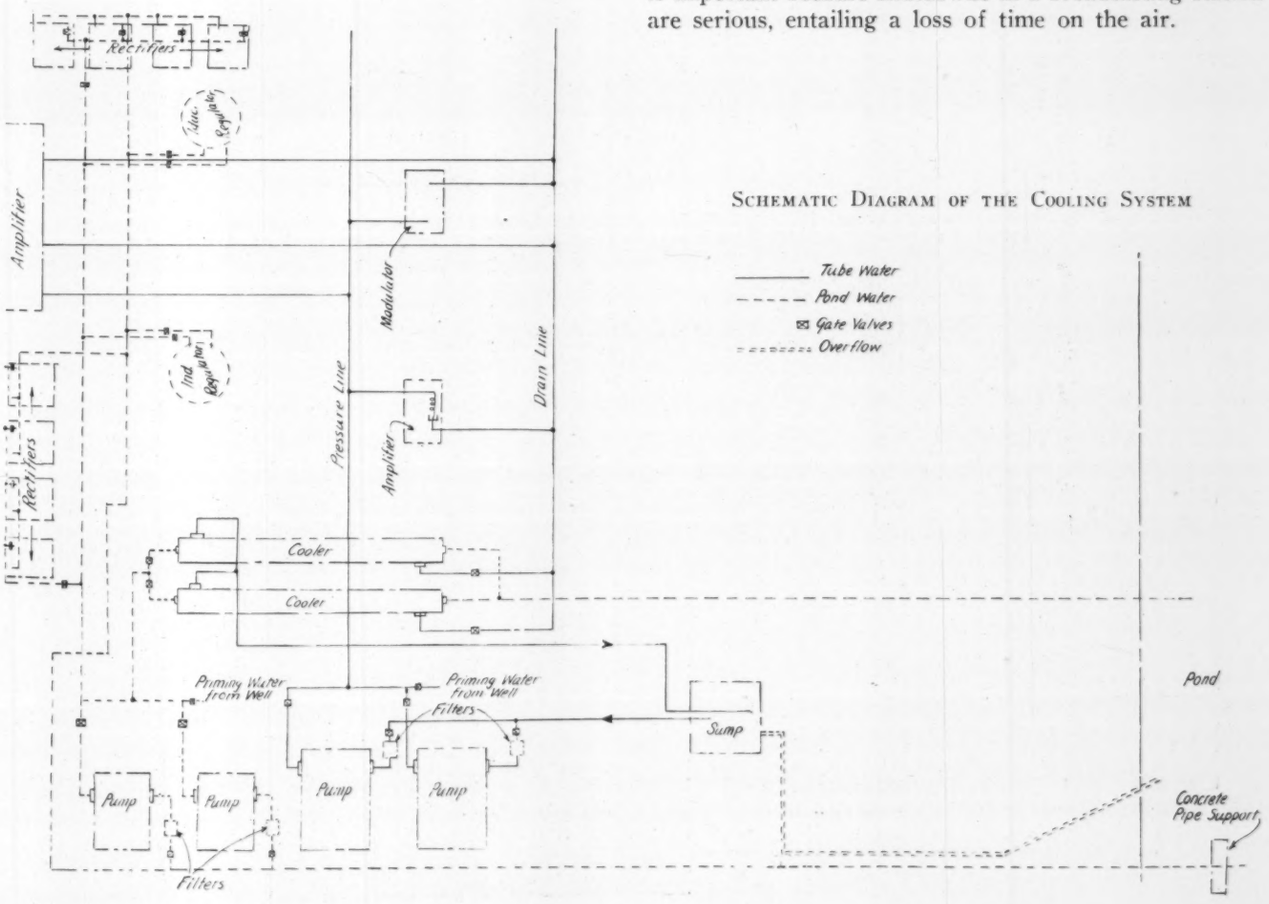
Water is obtained from this pond by means of a second pump system which forces the pond water up the interchanger and through the tubes. The water then flows back to the pond. Cooling is obtained at the pond by means of surface evaporation. The heat interchangers are mounted, from the ceiling, in the basement of the station, and, therefore, no danger from freezing is encountered from this source. To provide a further guard against freezing, the pipes to and from both the pond and the sump are laid underground a sufficient amount.

In addition to providing cooling for the power tubes, water cooling is also furnished for the rectifier systems and for the induction voltage regulators. These rectifier tubes are immersed in an oil bath, which, in turn, is cooled by means of the water system. This also applies to the induction voltage regulators which are in an oil bath with circulating water coils. The water for this system is obtained from the pond, and utilizes either the pressure drop across the heat interchangers or pressure obtained from a small auxiliary pump.

Each of the 100-kw. tubes has 25 gallons of water passing through the water jacket surrounding the plate of the tube and in addition to this amount, 2 gallons is passed through the center of the tube to cool the grid structure, thus making a total of 27 gallons per tube per minute. The smaller 20-kw. tubes use about 8 to 10 gallons of cooling water.

Incoming water from the sump has a temperature range from 30 C to 50 C and the water temperature after passing through the tubes ranges from 100 to 140 F. The cooling water from the pond ranges from 12 to 32 C. The above temperature ranges correspond to the winter and summer operating conditions respectively. The temperature of the voltage regulators is kept at or below 40 C, while the temperature of the rectifier tanks is kept at or below 32 C.

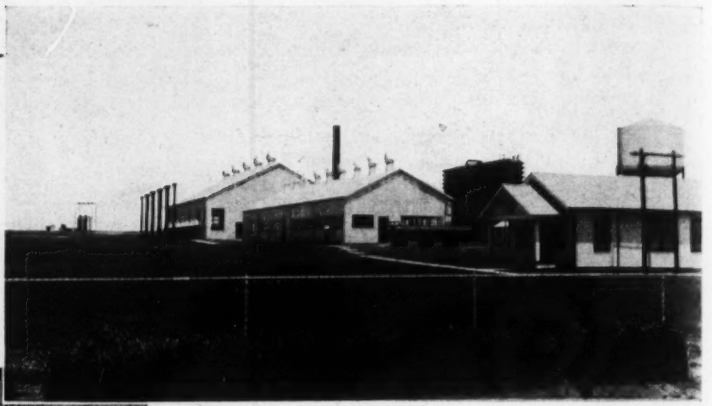
The water system has been installed in duplicate, two pumps being placed in each system and two heat interchangers provided. Valves are inserted in the system to enable cutting off any part so that work may be done on it without shutting down the whole system. This is important because shutdowns in a broadcasting station are serious, entailing a loss of time on the air.



SCHEMATIC DIAGRAM OF THE COOLING SYSTEM

Tube Water
Pond Water
Gate Valves
Overflow

The New 950-Mile Line from Texas



Side view of one of the ten compressor stations

Power end of main units in one of the compressor stations

Rear view of compressor station

The new natural gas line from the Panhandle field in Texas to Chicago is 950 miles in length in addition to several hundred miles of smaller diameter pipe in the lateral lines and gathering system. The entire main line and part of the gathering system is 24 inches in diameter. Two kinds of pipe are used in the main line; one has a wall thickness of $\frac{5}{16}$ in. and the other has an $\frac{11}{32}$ in. wall. The main line contains a valve about every ten miles. Coupings are about 80 feet apart.

Front view of employes' cottages at one of the compressor stations

Regulator house and water tank

Pipe Machine

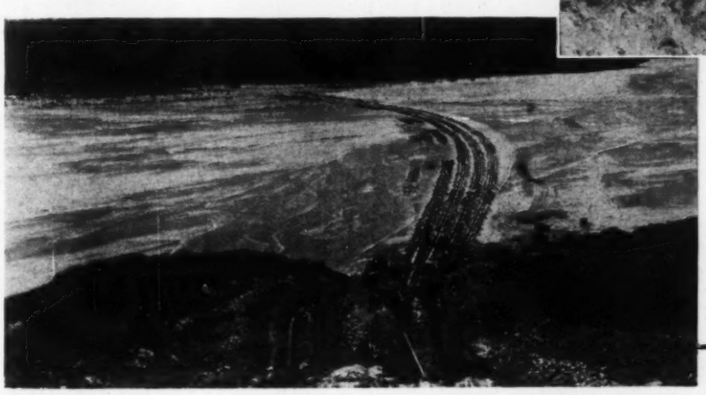
Coating 24-in. Pipe

Natural Gas Pipe to Chicago, Ill.

Line being lowered

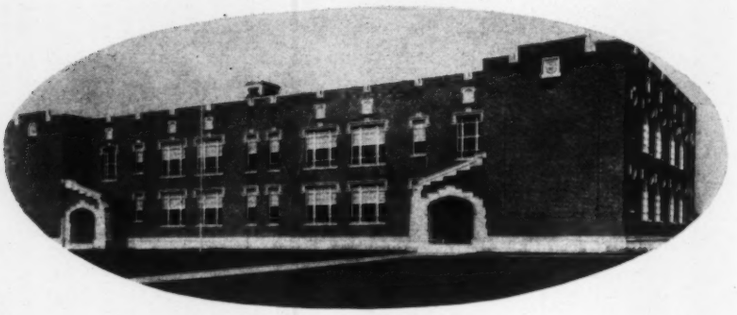
Natural gas will be mixed with manufactured gas to give a mixture of 800 Btu per cubic foot for industrial and heating use; gas will be charged for on the basis of therms—100,000 Btu. For heating gas, the rate schedule takes into account the amount of radiation the customer has installed. Among the rate classifications are schedules for the following services: Space heating and large volume water heating; water heating for off-peak use, May to Sept.; industrial; "interruptible" industrial; house heating; a special world's fair rate for operating exhibits.

Crane excavating on north bank of the Solomon River



Looking from north bank of Canadian River showing six completed 12-in. lines before being back-filled

Samuel R. Lewis*



THE BURROUGHS ELEMENTARY SCHOOL AT TOLEDO, OHIO. EDWIN M. GEE, ARCHITECT; WILLIAM HALLAUER, STRUCTURAL ENGINEER; SAMUEL R. LEWIS, HEATING AND VENTILATING ENGINEER

—describes the heating and ventilating of a twelve-room school. It has many interesting features

Method of Heating and Ventilating a Small School

THE Burroughs school is an example of the many years' experience of the construction and operating staff of the Toledo board of education. Edwin M. Gee is the architect of the board.

This staff has designed and built five very large high schools and about forty grade schools, these usually containing thirty-six classrooms, each with an auditorium, a gymnasim and various auxiliary playrooms, shops, etc. Many of these buildings are adapted to use, or are being used, as junior high schools.

It is rather refreshing, after a course of these larger projects, to try one's hand at a twelve-classroom building with an auditorium or gymnasium, like the Burroughs School.

Construction of Building

The building is of fireproof construction, without any basement except as required for the mechanical equipment. It is designed in such manner as to facilitate eventually doubling its size.

Since we know that no schoolroom floor ought to be directly upon the ground, a space high enough to work in is left between the bottoms of the first story concrete joists and the ground surface. The space under the corridor floor is excavated sufficiently to form a walk-through masonry air supply tunnel, and the steel duct risers to the various rooms convey the air to them from this tunnel.

There are two entirely separate supply fans each with its cold air intake and distributing ducts, one for the combination auditorium-gymnasium, and one for the classrooms. There is an electric-driven exhaust fan for the toilets.

Radiators Above Air Supply Openings

A feature of the plant lies in the fact that all air inlets to the rooms are along the floor against the exterior walls. The radiators in the classrooms are placed above these air supply openings, so that the entering air rises over and around the radiators, increasing their output.

The spaces between the reinforced concrete joists are used as air passages from the duct risers along the corridor walls to the room inlet-grilles along the outside walls.

* Consulting engineer, Chicago, Ill.

Steel Plates form Duct Bottoms

In order that the joist air-ways shall be smooth on all sides, anchor bolts or inserts are cast into the joist-bottoms, and flat galvanized steel plates reinforced with galvanized structural steel angles are bolted in place to form duct-bottoms. These plates do not interfere with the wire lath and plaster used for the ceilings of the rooms under them, and they are exposed in the shallow working space under the first story classroom floors.

The duct risers which serve the second story classrooms, being of metal, can be diverted one way or another in the first story height as convenient to coincide with the proper joist-spaces of the second story.

The recesses in the walls required for supply ducts to one room may serve through the next story above, also as vent duct spaces for the same room, since they have finished their "air supply duty" when they have fed air to the proper joist-spaces.

Air Distribution in Classrooms

The air distribution within each classroom seems to be especially effective with this method of introduction. The air emerges vertically along the wall in a thin sheet, and is then diverted and mixed with the down-coming cooler air from the glass and exposed wall. The mixed air turns out across the room in a refreshing but positive movement toward the outlets, which usually are at the floor along the inner walls of the room or at the inner ends of the coat rooms.

The Auditorum-Gymnasium

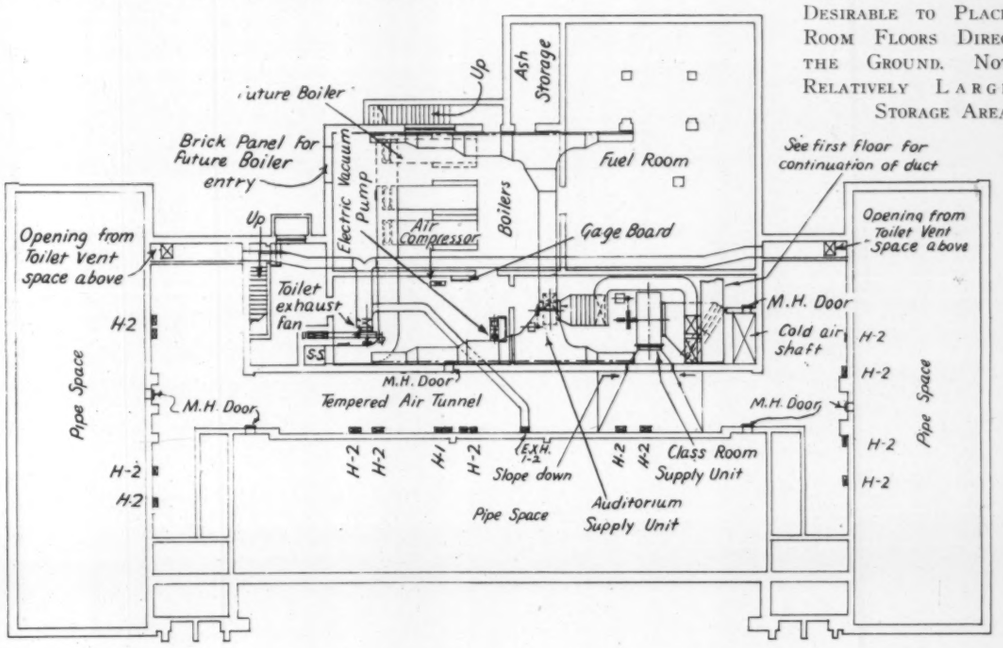
The auditorium-gymnasium receives its air supply along both sides through baseboard grilles, but does not have direct radiators above the grilles as in the classrooms.

The heating of the auditorium-gymnasium, therefore, is entirely indirect, with the radiators in the supply ducts under the floor. Arrangements are made so that all or part of the air in both the auditorium-gymnasium and in the classrooms may be recirculated, by manipulation of heavy single-leaf winch-operated dampers.

Stokers Will Increase Boiler Capacity

Steam is supplied by two down-draft soft-coal-burning self-contained firebox boilers. Space is left for a possible third similar boiler, and chimney capacity is allowed for it.

It will be possible, also, to increase the boiler output whenever desired by installing electrically-operated forced draft stokers. In one school of this type these stokers were placed during an emergency of short duration for the guaranteed development of 200 per cent of the original boiler rating.



Basement Plan. The Pipe Spaces Are Unexcavated, Being About Three Feet High. It Is Not Usually Desirable to Place Class Room Floors Directly on the Ground. Note the Relatively Large Fuel Storage Area

Steam at or Below Atmospheric Pressure

Circulation of steam is at atmospheric pressure or lower, and as part of the tempering heaters are lower than the boiler water line, an electric centrifugal vacuum pump is installed. This is arranged so that the water runs to its receiver without suction-lifts, and so that air from the interior of the piping has an untrapped flow to the receiver.

The tempering heaters are of cast iron, carried on structural steel. They are superimposed in this case due to exigencies of construction. This superimposing is usually not a desirable method of setting, since repairs are awkward in case of broken lower-tier sections.

Control of Steam Supply

No bypass dampers are provided around the tempering heaters and all sections are controlled by pneumatically-operated steam supply valves. This is contrary to practice elsewhere, but during many years of use it has caused no trouble in Toledo in plants having vacuum pumps.

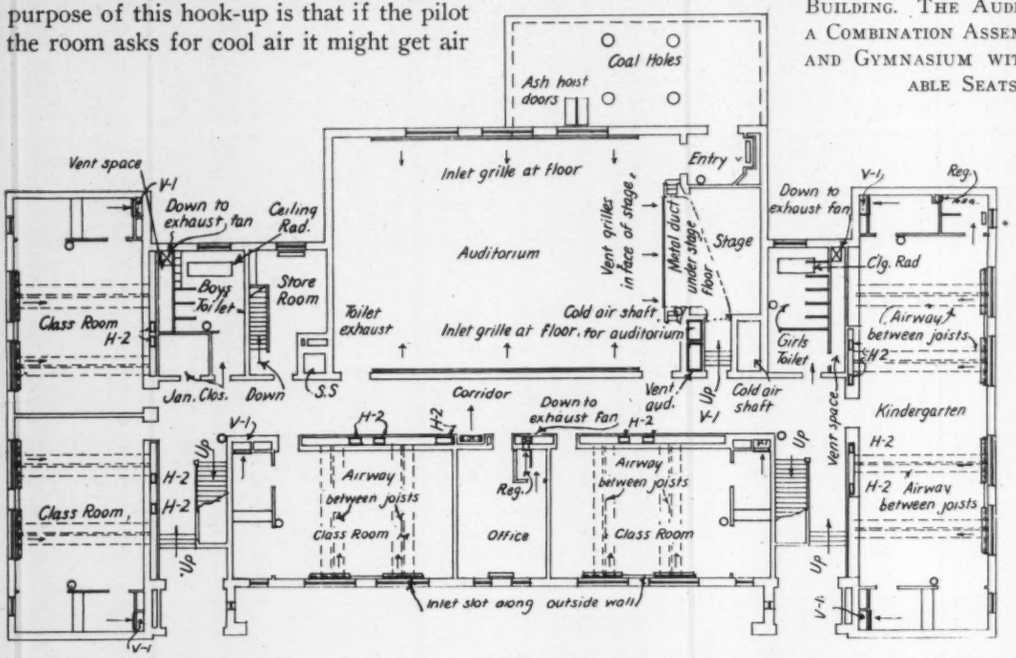
For the classroom supply system the three steam valves on the tempering heaters are controlled by a single three-stage duct thermostat in the supply fan discharge duct.

For the auditorium-gymnasium the five layers of heater are controlled by three supply valves, and these valves are controlled by a duct thermostat in the fan outlet. This thermostat, however, receives its supply of air, upon which its operation depends, from a pilot thermostat in the coldest part of the auditorium-gymnasium. The purpose of this hook-up is that if the pilot thermostat in the room asks for cool air it might get air so cool, relative to the temperature of the perhaps overcrowded room, that draft-consciousness on the part of the room-occupants would ensue. The duct thermostat stands like a watchdog to prevent this, since when it receives air from the pilot thermostat it goes immediately to work to limit the temperature of the air which passes it above a predetermined point at which the duct thermostat is adjusted to function.

Arrangement of Radiators in Toilet Rooms

In accordance with the long-established custom for elementary schools, the toilet rooms in the Burroughs school have no doors from the corridors, but are provided with over-lapping vision-screens. The toilet room radiators in most Toledo schools are suspended near the ceilings to facilitate cleaning the floors and walls.

FIRST STORY PLAN. THIS IS ABOUT HALF OF THE ULTIMATE BUILDING. THE AUDITORIUM IS A COMBINATION ASSEMBLY ROOM AND GYMNASIUM WITH REMOVABLE SEATS



The risers to these radiators are run inside the utility spaces behind the toilet fixtures, so that the latter are protected against freezing.

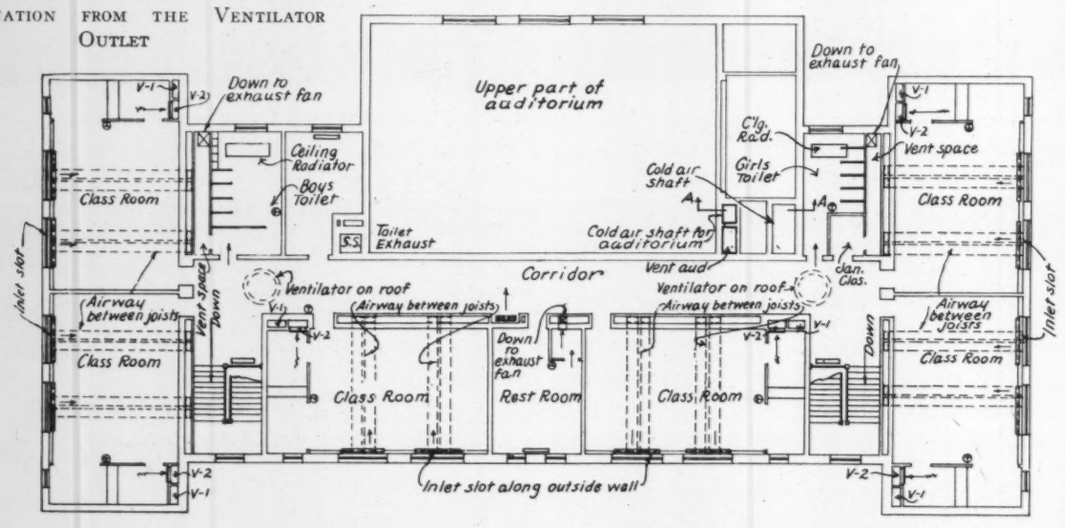
Location of Classroom Thermostats

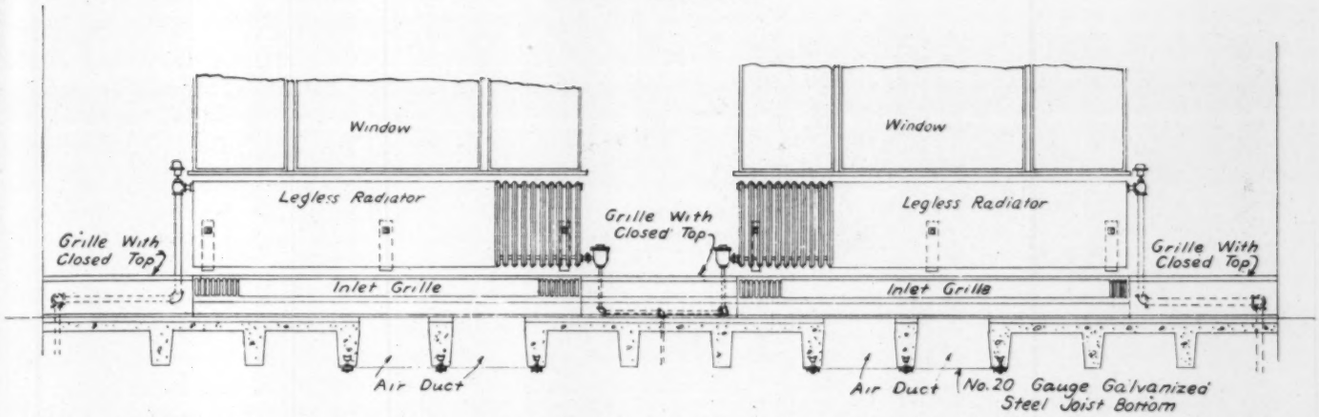
The classroom thermostats are placed on the interior side of the outer cased opening between the classroom and its coat room, this location having been established by experiment. No grilles are used on the vent openings from the coat rooms, the elbows being finished smooth, and painted to match the rooms. Thus they must be swept out at each cleaning of the floor.

Air Intakes

Cold air is taken into the building high above the dust-collecting ground, yet not directly above any similar

SECOND STORY PLAN. THERE IS A FIREPROOF EXHAUST CHAMBER ABOVE THE CORRIDOR WHICH LEADS TO THE TWO ROOF VENTILATORS. THE COLD AIR INTAKES ABOVE THE ROOF ARE ARRANGED SO THAT THERE IS NO CROSS-CONTAMINATION FROM THE VENTILATOR OUTLET





TYPICAL ELEVATION, SHOWING AIR SUPPLY FOR CLASS ROOMS. THE SUSPENDED RADIATORS ARE ABOVE THE CAST IRON BASEBOARD GRILLES THROUGH WHICH THE AIR SUPPLY ENTERS THE ROOM. THE PIPE RUN-OUTS TO THE RADIATORS ARE HIDDEN BEHIND THE METAL BASEBOARD. THE AIR SUPPLY CROSSES THE ROOM THROUGH THE SPACES BETWEEN THE CONCRETE JOISTS

roof-surfaces. Experience seems to show that if exhausted air is delivered through roof ventilators an excellent place for air-intake is high-up through the face of a vertical wall, as just below the eaves of a building.

It is desirable to have the exhaust outlet higher than the cold air intake as the relatively warmer discharged air will tend to rise above the unheated incoming air and no cross-contamination seems ever to ensue.

Where both inlet and outlet are above the roof it is desirable to have them as close to each other as possible, the outlet looking straight up, the inlet well below the outlet and looking out sideways.

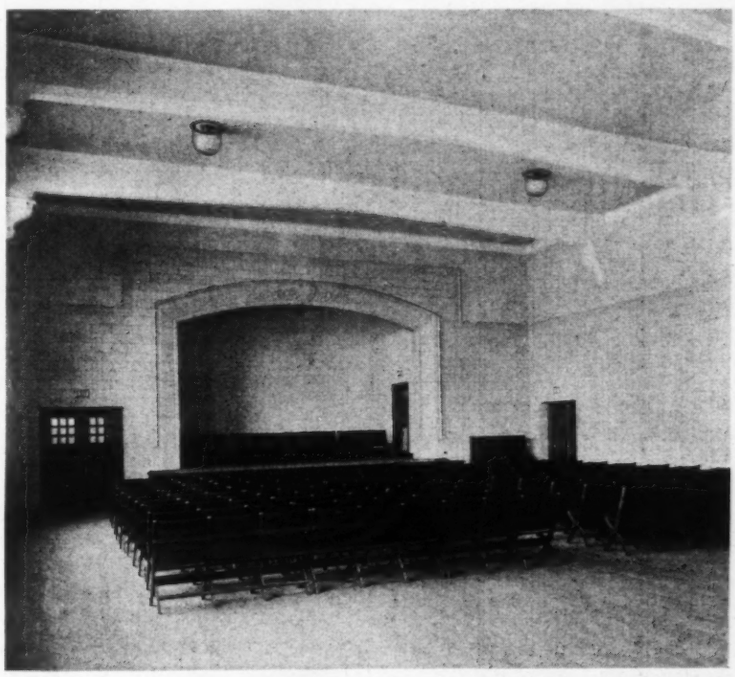
The heating and ventilating plant in the Burroughs school is typical of a number of older and larger school systems—six in fact—using radiators mounted above the air supply inlets, which have served satisfactorily during several heating seasons. These older plants, all essentially like the Burroughs' system, were developed after still earlier plants in which but one or two rooms, perhaps, were served experimentally by inlet-slots along the outside walls.

Cost of the Plant

The contract value of the fireproof Burroughs school (let in October 1929) is $160,700.00. Based on the standard cubing formula of the *American Institute of Architects* this gives a cost of 28.7 cents per cubic foot.

The heating, ventilating and automatic temperature regulation contracts, included in the foregoing sum, were let for $20,895.00, or 13 per cent of the total, or 3.73 cents per cubic foot.

Schools equipped with slot air-inlets are also in operation in Columbus, Ohio; Oak Park, Ill.; Bloomington, Ill.; and at Center Township, LaPorte, Ind. A number of additional buildings using it are in course of construction in Ohio and in Indiana.



INTERIOR OF THE AUDITORIUM. THE AIR INLET GRILLE IS TO THE LEFT ALONG THE WALL UNDER THE WINDOW. EXHAUST AIR LEAVES THE ROOM THROUGH GRILLES ACROSS THE FRONT OF THE STAGE

AN UNUSUAL VIEW OF THE ALL-STEEL-AND-GLASS BUILDING, WHICH IS HEATED BY A FORCED HOT WATER SYSTEM. FUEL CONSUMPTION WAS CONSIDERABLY UNDER THE ESTIMATED AMOUNT

By A. I. Snow*

An interesting arrangement is used to keep the water circulating at all times in this forced hot water system. A steam turbine connected to one end of the circulating pump is regulated to exhaust as much steam as is needed to maintain the desired water temperature, and a motor connected to the other end supplies the balance of the power to run the pump at the motor's synchronous speed.

All-Steel-and-Glass Building Proves Economical to Heat

AN all-steel-and-glass structure—artistically consistent with the beauty of the old masters' suits of armor, arms, and handiwork, and embodying the promise in the array of modern steel products—houses the John Woodman Higgins Armory collection at the Worcester Pressed Steel Company, Worcester, Mass.

Many objections arose when this type of building was proposed by Mr. Higgins, president and treasurer of the company. It was thought that such a building would be unstable, could not be made tight due to unequal expansion, could not be heated, would have frosty windows, and would rust rapidly. Finally, a garage was designed as an experiment. This was erected in 1927 and furnished the data needed for plans for the museum building.

In 1928 construction of the building was started by the Worcester Pressed Steel Company under the guidance of J. D. Leland & Company, architects, and Harty & Brown as managing engineers.

Forced Hot Water System Used

The building—whose outside walls are all steel and glass—is heated with a forced circulation hot water system, with which the remainder of the plant had been equipped some four years back.

The radiation is cast wall type or pipe coils about desk high in front of all the windows, to insure upward

* Plant Engineer, Worcester Pressed Steel Company, Worcester, Mass.

circulation and prevent cold down-drafts at the floor and at the desks placed along the windows. There are valves on the inlet and the outlet of each set of coils or radiators, but they are used only in rare instances or for repairs since the final system balance was made.

The museum, on the third floor—a lofty, vaulted Gothic room covering both wings—is heated to 50 F by coils along the glass sides, supplemented by a hot water unit type fan heater at each wing, thermostatically controlled to 70 F. This arrangement automatically takes care of variations due to sun or wind.

The temperature of the water for the whole plant is controlled according to the outside temperature as indicated on a recording thermometer in the boiler room.

The flow of water to the museum building is controlled mainly at the boiler room with the valve on the return line and regulated to maintain about 70 F near the office manager's desk. Any appreciable variation is 'phoned to the engineer, who makes the necessary water temperature correction.

The circulation pump is connected on the one end to a steam turbine whose greatest speed is about 1740 r.p.m. and water rate purposely high. The other end is connected to a 1750 r.p.m. alternating current motor controlled by the usual type of oil compensator, overload relays and low voltage coils.

In operation, the motor is always on the line and only as much steam put through the turbine at 100-lb. pressure as will provide enough exhaust steam to heat the circulating water by means of a closed tubular type heat interchanger operating up to 5-lb. back pressure.

Live or high pressure steam is added direct to interchanger through a reducing valve during colder weather to make up the steam requirement.

The greatest gain occurs at temperatures above freezing where only a part of the exhaust steam is needed and the motor picks up the load directly as the turbine decreases or increases its load in exact proportion to the heat needed.

This method of operating, put into practice just this past year to meet conditions due to greater head in the new building, provides more uniform temperature control in all parts of the plant due to maintained velocity through pipes, thus eliminating the dual necessity of trying to control velocity of water and temperature too.

The instrument panel contains a suction pressure gage on the pump intake and a discharge pressure gage on the outlet side. These are a check against proper flow conditions and detect improper use of zone control.

A chart showing best average water temperatures at prevailing outside temperatures is used with an outside recording thermometer and a discharge water temperature recorder to enable the engineer to control steam through the turbine to get the correct temperatures.

Thermometers on the return lines indicate which sections are in need of more or less heat, and the operating of valves on these return lines increases or decreases the volume of water forced through the different sections and thereby accurately controls the heat to these sections.

The use of the return valve seems to maintain better velocity pressure throughout the system, thus keeping the flow in the top and bottom of the building more uniform, especially during mild weather when slow circulation is needed and the water temperature can not be reduced on account of the remainder of the plant.

The two wings are also separately controlled by one return valve each and balanced so that the return water is at about the same temperature. Further control on the sunny sides is again obtained through divisional return valves. These are operated at about 10 a. m. and 3 p. m. to compensate for the 6 F average temperature increase due to the sun.

Practice has shown it best to keep the water circulating at all times rather than to stop the flow in the entire system, as this unbalances the temperatures of the different buildings due to different exposures. To accomplish this the steam turbine connected to one end of the circulating pump is regulated to exhaust as much steam as is needed to keep the water at the desired temperature, and the a-c. motor connected to the other end of the pump thereby supplies the balance of power to keep the pump up to the motor's synchronous speed.

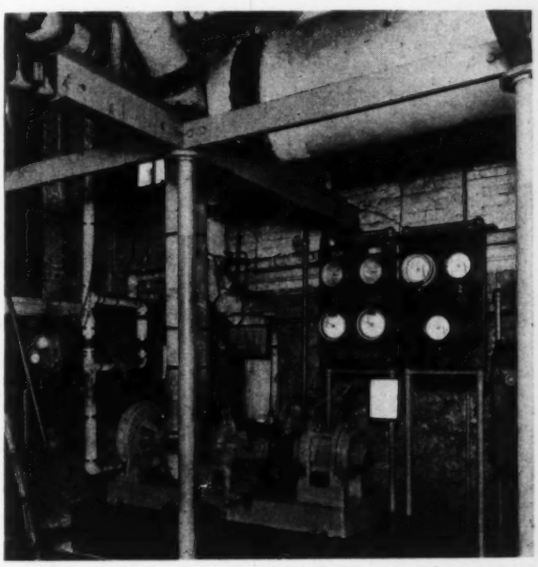
In this way only as much exhaust steam is used as is needed for heat and no current used that is not needed, with constant pressure maintained.

While there are no instruments for measuring the exact amount of coal used for this building, past records and estimates based on past performance established $2,090 as the probable cost of fuel for this building. Actual cost for last season was $1,195, even though a slightly lower average temperature was experienced.

The coal used was 225.5 tons more than before the new building was heated. This figure represents the actual increase in coal bought over previous years, and is therefore approximate.

It is believed that the tightness of the building, the carefully placed radiation, and the reasonably close central regulation account for the surprising comfort of this all-steel-and-glass building during the cold season.

Plans for humidity control are now under way based on the absence of frost on any of the glass during the coldest weather. After a very warm summer, it has been found that this building is extremely comfortable. About one-third of the windows are opened at 4 a. m. to allow cool air to remove heat from the offices.



THE CIRCULATION PUMP, MOTOR, TURBINE AND INSTRUMENT PANEL

Curves Aid in Design of Thick-Walled Tubes and Cylinders

With the curves presented here, the piping engineer is able to make use of the Clavarino formula in the design of thick-walled tubes and cylinders with minimum computation.

By F. E. Wertheim

WHEN hydraulic presses were first used, fractured cylinders were far from being unknown, and consequently cylinder walls were made heavier when cracked cylinders were replaced. Later, when it was found that increase of wall thickness beyond certain limits did not result in a corresponding increase in strength, investigation of the strength of thick-walled tubes and cylinders began. This subject was of such importance in connection with the design of guns that many attempts were made to originate a formula.

Among these formulas were those of Brix, given in the "Konstrukteur" by Prof. F. Realeaux; of Grashof, given in Unwin's "Machine Design"; of D. K. Clark, Lamé, Barlow, Birnie, and Clavarino. Burr and Lanza developed formulas giving results identical with the Lamé, and Merriman developed one producing results identical with Barlow's.

Inasmuch as the modified Barlow formula has found a place in the A. S. M. E. Boiler Code, this comparatively simple formula will continue to find ready acceptance. Its scope of usefulness is clearly defined in the recent article by Harvey A. Wagner. (HEATING, PIPING AND AIR CONDITIONING, August, 1931, p. 658.)

For figuring thick-walled tubes or cylinders subjected to internal strain, use of the Clavarino formula is good practice. In order to simplify the calculations for application of this formula, tables [1] are available which save time and mathematical work. Use of these tables is limited, however, because they do not apply for calculations in which the known quantities are inside diameter, working fiber stress, and working pressure.

Curves for Use with Clavarino Formula

The accompanying sets of curves, Fig. 1 and Fig. 2, have been devised for use in the solution of problems involving the Clavarino formula. In making the computations for these curves the form of the Clavarino formula given by Merriman was used. Merriman here assumes a value of 1/3 for Poisson's ratio, while the value now generally accepted is 0.3. Because these curves are intended to cover a wide range of application, and the Merriman form has been much used for important work, the writer has given it preference in the present instance. Investigations to determine Poisson's ratio accurately, especially at high temperatures, are promised [2] and should yield very useful information.

It will be noted that pressure ranges on these curves

[1] National Pipe Standards (1924).

[2] See "Design of Thick-Walled Tubes Subjected to Pressure and Heat Input" by E. W. Luster, p. 167—paper F. S. P.—53-12, Trans. *A. S. M. E.*

are from 1,000 to 15,000 lb. per sq. in. Except for ordnance, pressures of 15,000 lb. seem tremendous; nevertheless, machines to compress gas to such a final discharge pressure are in use in the nitrogen industry.[3]

The working fiber stresses given on these curves are intended for application to the maximum conditions now being considered in steam engineering. For 3,500 lb. W. S. P. at 1,000 F there is a probability of the development of a new alloy steel with a working fiber stress of not less than 14,000 lb.[4]

Example Shows Use of Curves

An example to show the use of these curves follows:

Given: Inside diameter = 16 in. ($r = 8$ in.)
Working fiber stress = 15,000 lb.
Pressure = 4,000 lb. per sq. in.

Referring to Fig. 1, $C = 0.3$
$t = r \times C = 8 \times 0.3 = 2.4$ in., say $2\frac{7}{16}$ in.
∴ Outside diameter $= 16 + 4\frac{7}{8} = 20\frac{7}{8}$ in.

The writer hopes that these curves may find their way into the note books of engineers and designers and believes that they will be of utility.

[3] See *Power*, p. 764, Nov. 11, 1930; and *Mechanical Engineering*, p. 511, July, 1931.

[4] See "Meeting on Steel Pipe Flanges and Flanged Fittings," *Mechanical Engineering*, p. 717, July, 1930.

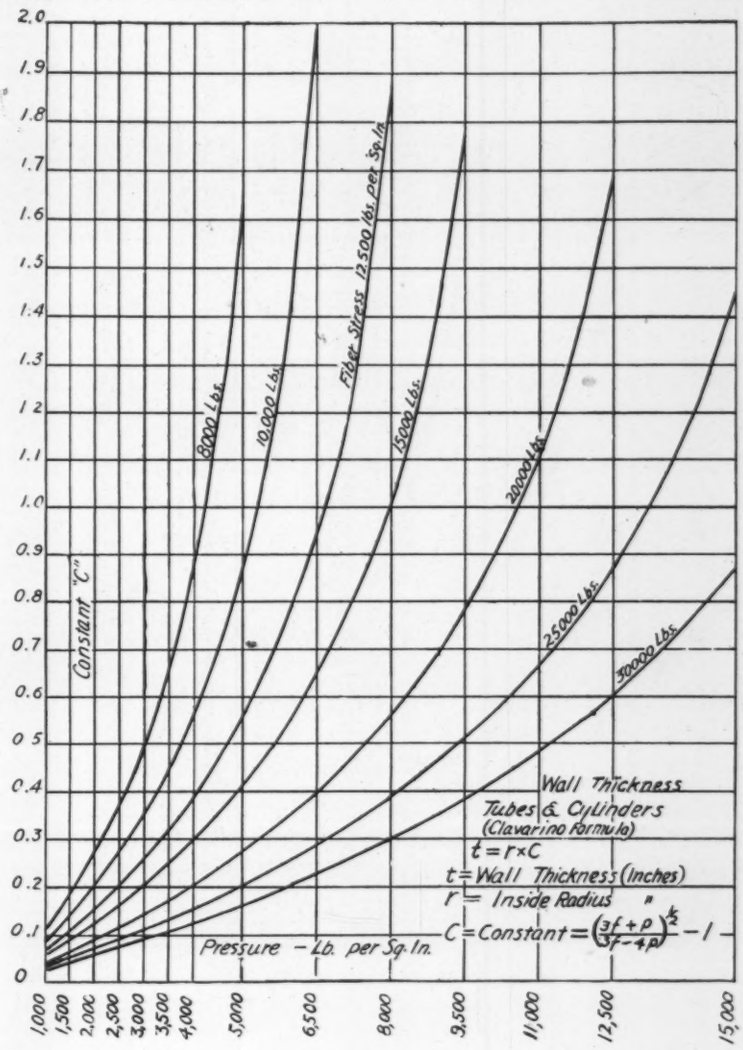

Constant "C"
8000 Lbs.
10,000 Lbs.
Fiber Stress 12,500 lbs. per Sq. In.
15000 Lbs.
20000 Lbs.
25000 Lbs.
30000 Lbs.
Wall Thickness
Tubes & Cylinders
(Clavarino Formula)
t = r × C
t = Wall Thickness (Inches)
r = Inside Radius "
C = Constant = (3f + p)/(3f - 4p) ^ 1/2 - 1
Pressure - Lb. per Sq. In.
1,000 1,500 2,000 2,500 3,000 3,500 4,000 5,000 6,500 8,000 9,500 11,000 12,500 15,000

Fig. 1

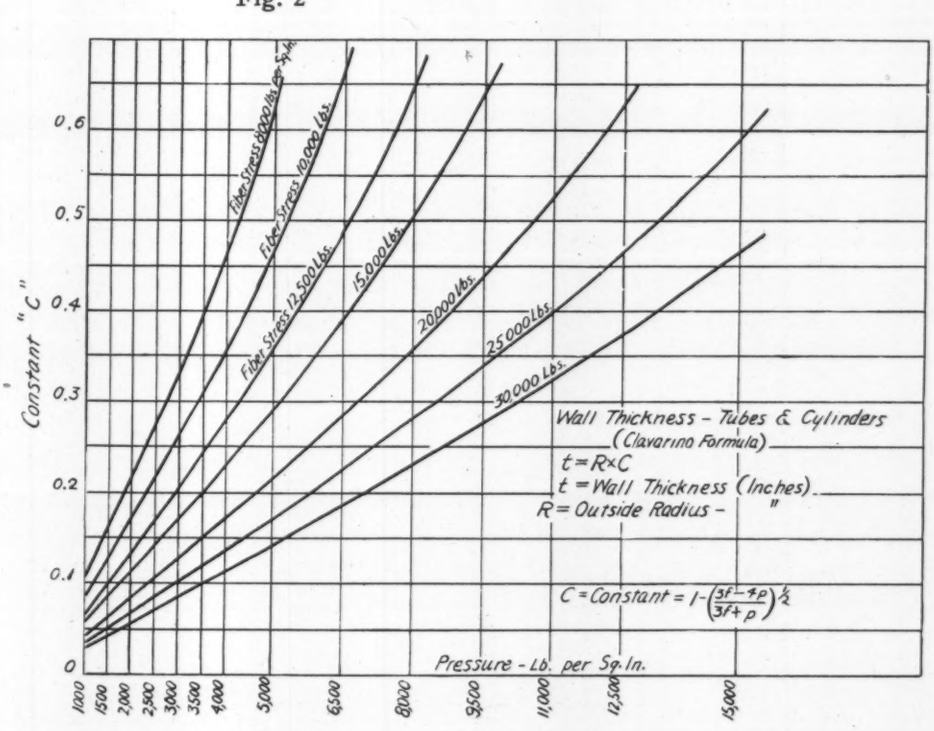
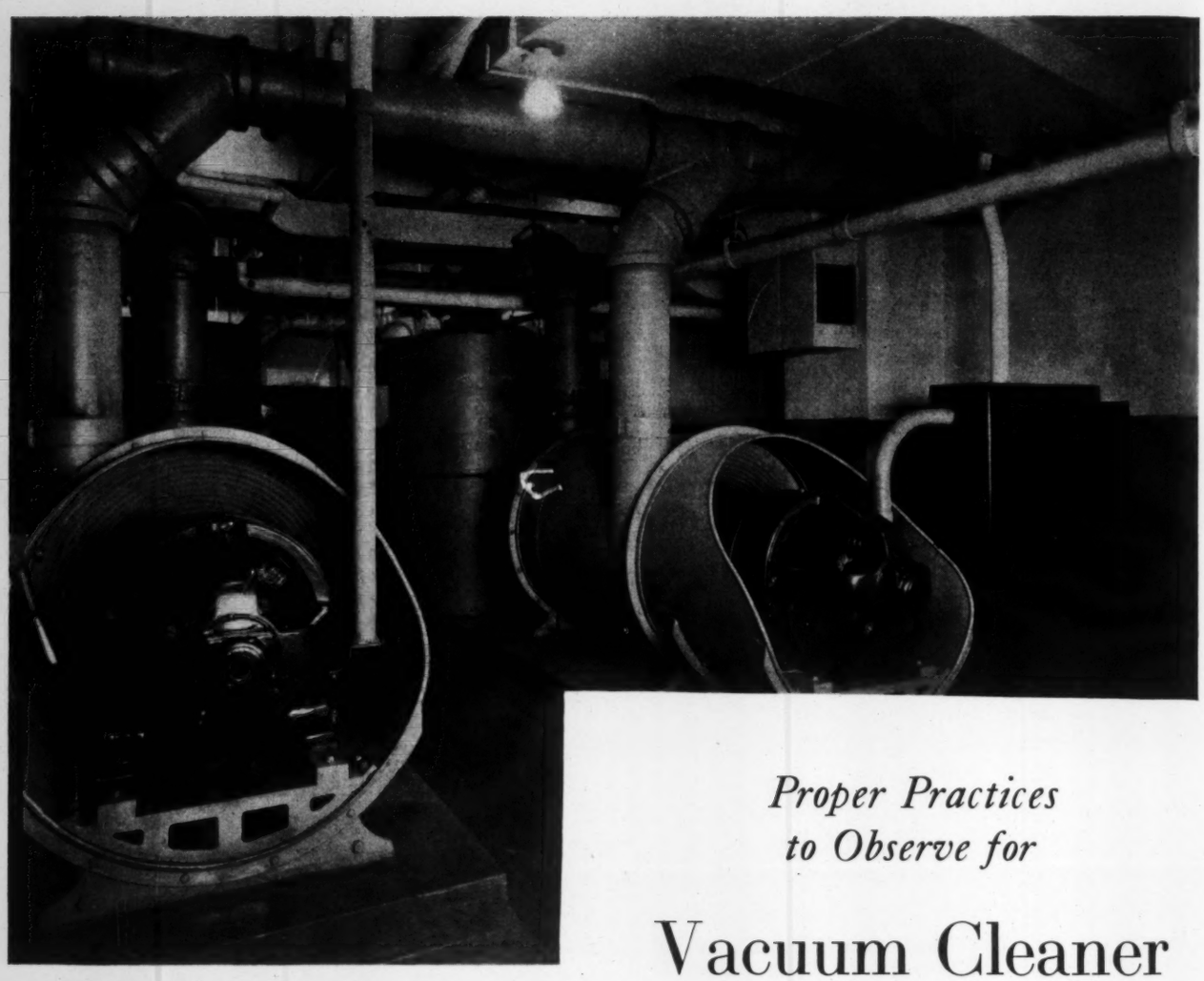

Constant "C"
Fiber Stress 8000 lbs per Sq. In.
Fiber Stress 10,000 Lbs.
Fiber Stress 12,500 Lbs.
15,000 Lbs.
20000 Lbs.
25000 Lbs.
30000 Lbs.
Wall Thickness - Tubes & Cylinders
(Clavarino Formula)
t = R × C
t = Wall Thickness (Inches)
R = Outside Radius - "
C = Constant = 1 - (3f - 4p)/(3f + p) ^ 1/2
Pressure - Lb. per Sq. In.
1000 1500 2000 2500 3000 3500 4000 5000 6500 8000 9500 11000 12500 15000

Fig. 2



CENTRAL VACUUM CLEANING SYSTEM INSTALLATION IN AN OFFICE BUILDING

Proper Practices to Observe for

Vacuum Cleaner Piping Systems

By Lester C. Smith*

IN CENTRAL vacuum cleaning systems in office buildings, schools, hotels and similar structures, the piping is of paramount importance. For the benefit of engineers who design, operate or maintain these systems, this article calls attention to some of the latest accepted practices in their installation and operation.

Advantages of Vacuum Cleaning Systems

Vacuum cleaning plants afford a means whereby buildings may be thoroughly cleaned in a shorter period of time and at less cost than can be done with broom and mop. For instance, the average amount of space cleaned with a broom is 1,000 sq. ft. per hr. whereas with a central vacuum cleaner one operator can clean about 1,400 sq. ft. per hr. The above figures are based not on cleaning only but on doing such related work as may be also necessary in an allotted space. For instance, there are waste-paper baskets to empty, occasional mopping in toilets and similar places and cuspidors to be cleaned. Accordingly, the above figures take this work into consideration as well as the actual sweeping involved. Where an operator is doing nothing but vacuum cleaning he can usually cover about 3,000 sq. ft. per hr., so on this basis, of course, space may be covered much faster than where incidental work is also taken into consideration. The advantages of vacuum cleaning systems result in a reduction in the number of necessary employes and a resulting saving in maintenance cost. It has been shown on installations which have been in operation for a considerable period of time that besides paying the interest and depreciation charge on the additional investment, a central vacuum cleaning system shows an appreciable decrease in yearly maintenance costs.

Ratings—Sweeper Capacity

It may be of interest to explain briefly the ratings which are given on central vacuum cleaning systems. These equipments are rated by what is known as *sweeper capacity*. In other words, they are designed to operate one or more cleaning implements simultaneously. In figuring the motor load for any given installation, 2½ h.p. are allowed for each sweeper of capacity that the

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plant has. Accordingly, therefore, a four-sweeper installation requires a 10 h.p. motor to operate or drive the vacuum producer.

Before installing a system in any building it should be first determined how many operators would be necessary to clean the entire structure in the time which will be allotted for this purpose. On the basis that a given operator can clean 3,000 sq. ft. per hr. one should first determine the total number of cleaning hours which will be available each day. Then by dividing the total number of sq. ft. area in the entire building by the space each operator should clean in this time, the answer will indicate the number of sweepers of capacity that the vacuum plant should have.

For example, consider a building of 500,000 sq. ft. floor area, with seven hours available each night for vacuum cleaning. One operator, if cleaning all of the time, will cover 21,000 sq. ft. Thus, the vacuum system should have sufficient capacity to permit twenty-four operators working simultaneously. This does not mean that twenty-four employes will keep this particular building in first class condition, since there is other incidental work aside from regular cleaning. The above figures, therefore, should be used merely in estimating the capacity of the vacuum producer to be installed. To determine the number of employes actually required in this building, 1,400 sq. ft. per hr., which includes vacuum cleaning and incidental work, should be used. Thus it is found that a total of about fifty-one employes would be needed to maintain the entire building doing all work, such as dusting and mopping as well as the vacuum cleaning. The reason that a twenty-four sweeper vacuum plant is sufficient to accommodate fifty-one operators is due to the fact that a load factor enters here, with the result that probably never more than half of the workers are vacuum cleaning at any one time. Thus a twenty-four sweeper plant takes care of the installation very nicely, although practically twice this number of employes are needed for the cleaning of the entire building.

It has developed in a number of buildings where the vacuum cleaning system has been installed that it is advantageous to have the operators work in pairs. Thus, for instance, two women are assigned a certain amount of floor space which is under their entire care. While one woman is vacuum cleaning this area, the other woman is occupied emptying waste-paper baskets, cleaning cuspidors, dusting and doing the other incidental work which is necessary.

Designing the Piping

A carefully designed and constructed vacuum cleaning system is useless unless the piping system on which it operates is also carefully designed and of the proper proportion for the work which is to be done. It is best to plan the installation of a central vacuum cleaning system at the time a building is erected. The piping system is, of course, fairly extensive and each installation can be better handled before the walls and floors are finished than afterwards. However, in numerous instances a central vacuum system has been installed many years later as part of a general rebuilding program, for instance.

To secure permanent efficiency, piping of ample size with outlets at easily accessible points is quite necessary. The piping should not be so small as to require too high a velocity with consequent frictional loss in vacuum as well as possible sand-blast wear. On the other hand, it is essential that the piping should not be excessive in size, as this means a reduction in the air velocity which results in dust settling in the horizontal pipe lines connecting all risers. Furthermore, a system which is intended to handle dirt-laden air must be of full uniform bore throughout both piping and fittings. Long turn

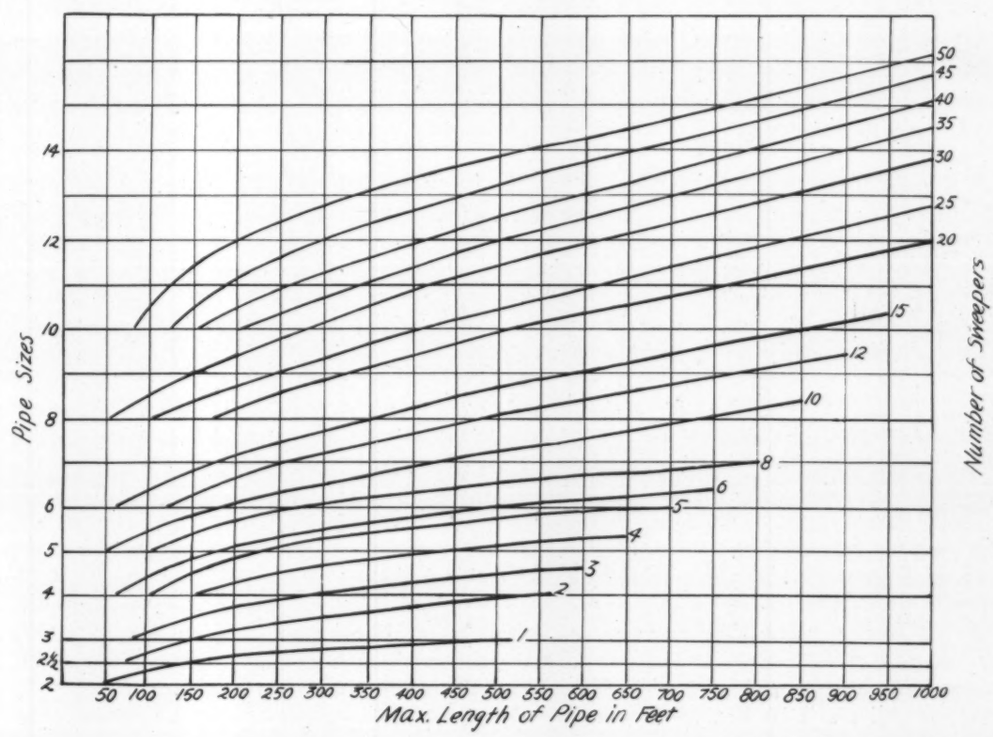


FIG. 1 — SWEEPER CHART

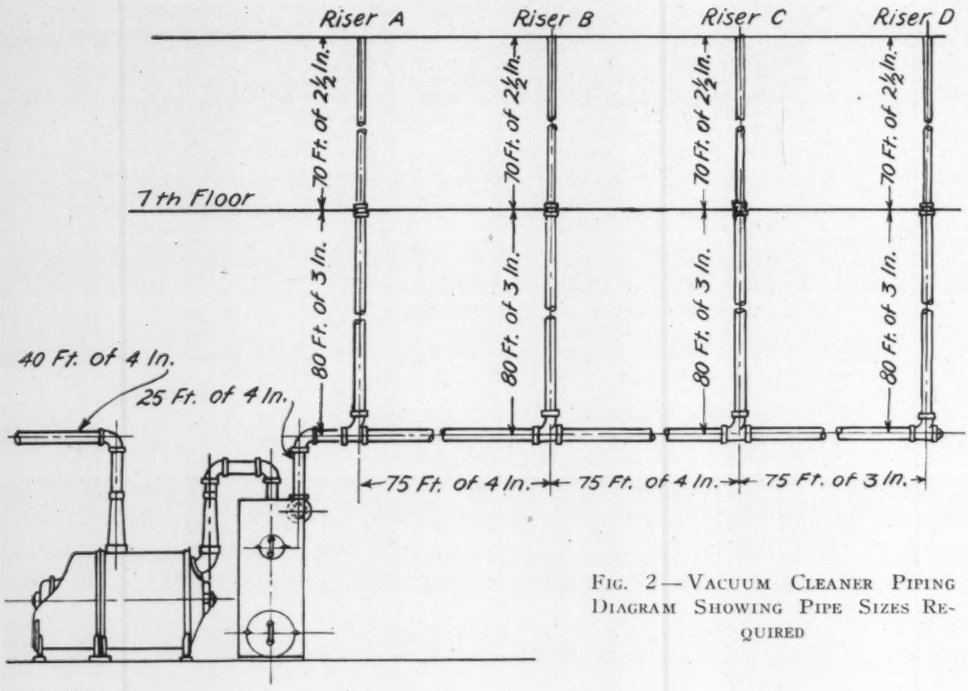


FIG. 2—VACUUM CLEANER PIPING DIAGRAM SHOWING PIPE SIZES REQUIRED

drainage type fittings have been found to be satisfactory for this application. Care, however, should be exercised in selecting the correct fittings for each location.

Charts Available for Sizing

There are various charts and schedules available for use in the selection of piping sizes for any installation. These charts vary somewhat in general appearance but if they are properly used the results are approximately the same in most instances. Nothing smaller than 2-in. pipe should be used anywhere in a central system, and this size should be used for only short distances where one sweeper is to be operated on a particular branch. Furthermore, in all public buildings, schools, offices, hotels, etc., where matches and similar litter might be picked up, 2½ in. pipe should be regarded as the absolute minimum regardless of height or distance conditions. This is due to the increased liability of clogs in the smaller fittings from such particles as matches, hairpins and other things.

Another point to be borne in mind is that a standard practice requires that the piping system be laid out so that any part of the structure may be reached with a 50-ft. length of vacuum hose. In other words, the vacuum inlets should be located at such frequent intervals that any point in the building can be reached with 50 ft. of hose connected to some one of these valves. The first thing in laying out a piping installation is to locate these valves roughly on the various floors, rooms, corridors and stairways so that all parts of the building are covered. On the different floors, these valves should, of course, be so located that they may be all connected to one riser coming up from the basement. As a general rule, buildings have a similar floor plan throughout so this is usually not difficult. Thus, a given riser may run from the top of the building to the basement and have a vacuum inlet on every floor. By locating a sufficient number of these vacuum risers it is possible to provide an ample number of inlet valves throughout the entire structure.

The next step is to assume—as nearly as possible—the greatest number of operators that may be connected to one riser at any given time. Thus, if there are several groups operating on different floors it is possible that three, four or possibly more operators may be connected to one riser. Fig. 1 is a chart which has been successfully used for laying out vacuum piping installations.

Fig. 2 shows a possible piping installation which is based on a four-sweeper plant with not more than two sweepers in use simultaneously on any one riser and not more than one sweeper at any one time above the seventh floor on any given riser. To determine the size of pipe required it should first be noted that the total distance from the machine to the most remote inlet valve is 400 feet. On the last seventy feet of this pipe line, however, only one sweeper is to be in operation, so by referring to the diagram we find that the one-sweeper curve intersects the 70-ft. line about opposite the 2½-in. pipe figure; consequently 2½ in. pipe should be installed for the uppermost seventy feet of this riser. For the next 155 feet back towards the machine, or back to where riser *C* intersects the main pipe line, two sweepers may be in use simultaneously at any one time. Then by referring to the two-sweeper curve where it intersects the 155-ft. line, it is found that a 3-in. pipe should be installed for the next 155 ft. back towards the vacuum producer. Since the operation has been assumed to be identically the same on risers *A, B, C,* and *D,* the same procedure may be used for figuring these other risers and the result is, of course, that for the uppermost seventy feet, 2½-in. pipe is satisfactory whereas at the bottom, eighty feet of 3-in. pipe are required.

On the horizontal main, from the place where riser *C* intersects back to the machine, it is possible for four sweepers to be operating at any one time; consequently,

this problem must be solved separately. This distance amounts to 175 ft., so if the four-sweeper curve is followed to where it intersects the 175-ft. line, it will be noted that a 4-in. pipe is required. Accordingly, a 4-in. pipe should be used back to the separator and from the separator to the machine.

For the vacuum exhaust connection, which should be

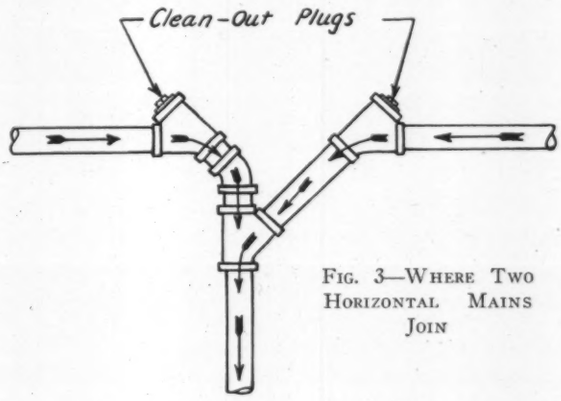


FIG. 3—WHERE TWO HORIZONTAL MAINS JOIN

made to stack, it is customary practice to install either the same size pipe as the largest on the intake side of the machine or the next size larger. This depends on whether the distance from the vacuum exhaust to the stack is long or short. On this particular diagram it is only 40 ft. so 4-in. pipe (the same as in the horizontal main on the other side) is shown. If the run were longer, 5-in. pipe would probably be used.

Figs. 3 to 7 illustrate several proper and also several improper means of using fittings. A careful study of the various drawings will show how these long turn drainage type fittings are used. In addition, there is illustrated in Fig. 8 a typical piping installation in a small building. In this drawing there is an attempt to approximate most of the usual conditions which are encountered. For instance, it will be noted that both concealed and open piping, the installation of wall valves and floor valves as well as the installation of short branch connections in the basement, are shown. Too much attention cannot be paid to the proper installation of these fittings.

Judgment Needed

A certain amount of judgment must necessarily be used in laying out any system. Structural peculiarities of individual buildings must be taken into consideration and often times these peculiarities require a certain departure from the usual procedure. Standard practice specifies that there should be a drop in suction of not more than one inch of mercury between the most remote inlet valve and the vacuum producer. It is well to check the pipe sizes against the resistance to flow after the piping system has been laid out. Table 1 gives the vacuum loss in inches of mercury for 100 feet of straight pipe; if the volume which is to be carried by any pipe is known the vacuum drop through the entire system or any part of it can easily be figured. When estimating "elbow loss" in a vacuum piping system it is customary to consider each elbow as equivalent to 10 feet of straight pipe. Accordingly, when figuring any given run of piping this figure should be added for each elbow that is encountered in that part of the system under consideration.

A volume of 70 cu. ft. per minute is figured for each sweeper which the system is to operate. This figure covers maximum conditions, or those conditions which are encountered in bare floor cleaning. In other words, the volume drawn through the tool in the vacuum cleaning of carpets is not as great as that volume drawn through the tool in vacuum cleaning bare floors or similar surfaces.

Below is published a typical set of piping specifications such as is usually given to the piping contractor where this work is done separately from the actual installation of the machine. These specifications are essentially the same, no matter what the make of the cleaner to be installed.

Piping Specifications for Cleaning Systems

The piping contractor shall furnish all necessary labor and material for the installation of all vacuum and exhaust piping required for the vacuum cleaning system. This system shall be installed in accordance with the piping layout as shown on the architect's or engineer's blueprints, which are to be considered a part of this specification. These blueprints show definite pipe sizes and locations and any proposed changes must be approved by the architect or engineer in writing.

The vacuum producer is to be supplied by another contractor but the piping contractor is to make all necessary piping connections to the machine. This shall be interpreted to include both vacuum and exhaust piping connections as well as the piping between the vacuum pro-

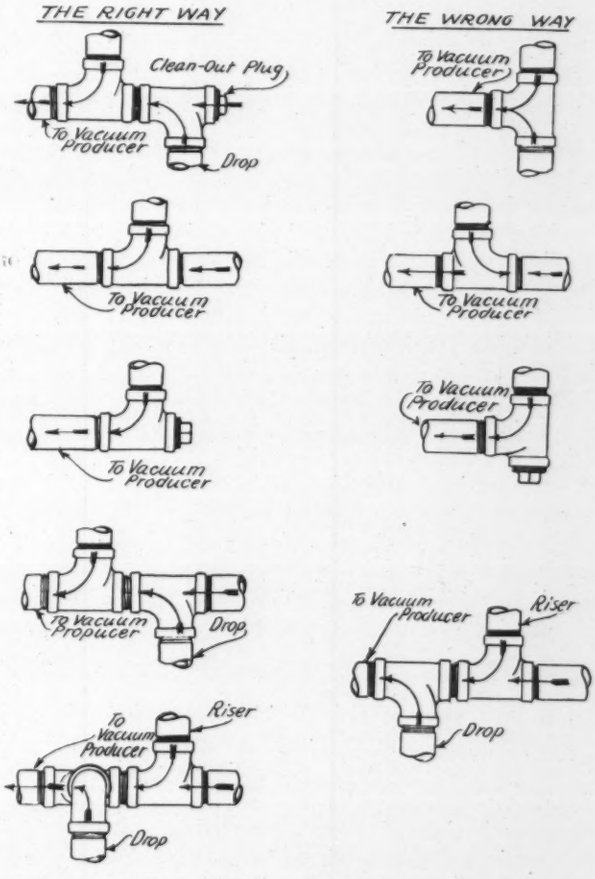


FIG. 4—RIGHT AND WRONG ARRANGEMENTS FOR VACUUM CLEANER PIPE FITTINGS

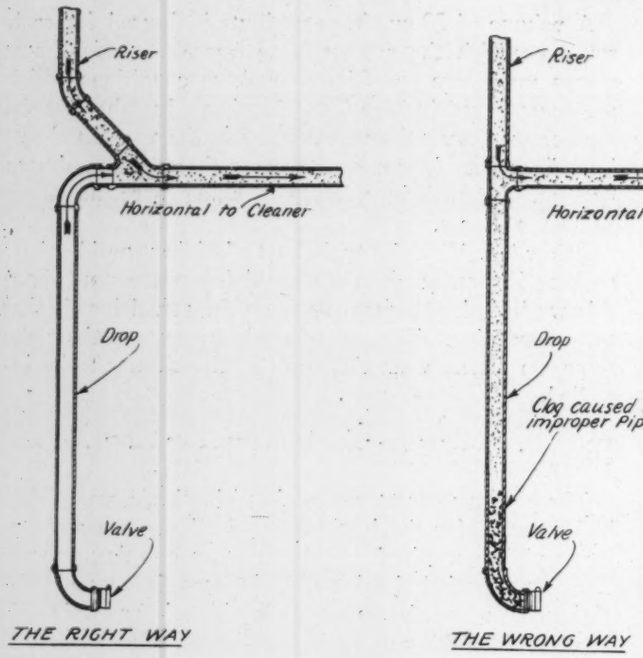


FIG. 5 — VACUUM CLEANER PIPING ARRANGEMENT

ducer and the dust separator. The piping contractor shall also be required to set the vacuum producer on a pad furnished by the manufacturer without fastening it to the floor in any way, and he should make the piping connections between the producer and the separator in accordance with the instruction sheet furnished by this manufacturer. The flexible sleeves furnished with the outfit must be placed next to the producer on both the intake and exhaust pipes. These connections must be straight and not installed at any angle of any kind.

All pipe used shall be of black mild steel or wrought iron as called for in the architect's or engineer's specifications. All pipes shall be cut square and reamed to the full inside diameter; all seams or projections removed and each connection screwed full home against the shoulder on the fittings. All pipe ends must butt where couplings are used, thus providing a smooth uniform bore throughout the entire system. All mains and risers should be run substantially as indicated on the plans.

All fittings used in connection with the vacuum piping shall be of the long sweep sanitary drainage type with recessed threads. Where available space prevents their installation, "Y" branch fittings are allowable but "T" fittings or 90 deg. elbows should be avoided. Steam, gas or water fittings, however, positively must not be used as they may clog.

Long radius pipe bends (similar to electrical conduit) should be used instead of elbows wherever possible. This especially applies where the horizontal mains are below the machine and the dirt has to be lifted against gravity to the separating tank. In the latter case a "T" with a clean-out plug should be placed next to the bend unless the flexible sleeve furnished with the machine is installed at this point.

Easily accessible, horizontally disposed clean-out plugs should be placed at the bottom of all risers and at the end of every horizontal run. These clean-outs should be screwed in so that they face in the direction of the air flow, thus eliminating any pockets for the accumulation of dust and dirt as it passes through the system.

Approved pipe hangers should be carefully installed on all horizontal piping and these should be located at intervals of 10 feet. All vertical piping should be rigidly braced so that the entire piping system is neatly installed in a thoroughly workmanlike manner.

Exposed piping where it passes through a finished floor or wall should be provided with nickel-plated floor or ceiling plates.

The exhaust pipe should be connected to a chimney or flue that has no other opening above it. If this exhaust pipe is connected to a stack which also serves boilers and other heating equipment, the exhaust outlet from the vacuum cleaner should be connected at a sufficient distance from the furnace connections so that at no time do flames pass the opening from the cleaner.

The piping contractor is to attach and install the service inlet valves at points indicated on the architect's or engineer's plans. These valves are to be provided by the manufacturer of the vacuum producer and delivered by him to the building. Wherever these valves are to be installed the end of the piping fitting should be left at such a distance from the face of the wall or baseboard

TABLE 1—VACUUM IN INCHES OF MERCURY LOST BY FRICTION IN PIPES 100 FEET LONG

Cu. Ft. Per Min.	1½ In.	2 In.	2½ In.	3 In.	4 In.	5 In.	6 In.	8 In.	10 In.	12 In.	14 In.
30	.37	.087	.029	.011	.003	.....	.....	.....	.....	.....	.....
40	.65	.160	.051	.021	.005	.0010	.....	.....	.....	.....	.....
50	1.02	.24	.079	.032	.008	.0020	.....	.....	.....	.....	.....
60	1.47	.35	.114	.045	.011	.0036	.0014	.....	.....	.....	.....
70	2.01	.48	.156	.063	.015	.0049	.0020	.....	.....	.....	.....
80	2.62	.62	.204	.082	.020	.0064	.0025	.....	.....	.....	.....
90	3.31	.89	.257	.104	.025	.0081	.003	.....	.....	.....	.....
100	4.09	.97	.318	.128	.031	.010	.004	.....	.....	.....	.....
150	.....	2.18	.715	.288	.069	.023	.009	.0016	.....	.....	.....
200	.....	3.73	1.39	.510	.120	.039	.016	.004	.....	.....	.....
300	.....	.....	2.90	1.14	.275	.089	.036	.008	.0028	.....	.....
400	.....	.....	5.09	2.05	.489	.158	.064	.015	.005	.....	.....
500	.....	.....	.....	3.20	.775	.250	.100	.024	.008	.0032	.....
600	.....	.....	.....	4.60	1.10	.357	.145	.034	.011	.0046	.....
700	.....	.....	.....	.....	1.52	.490	.196	.046	.015	.006	.0029
800	.....	.....	.....	.....	1.85	.634	.258	.061	.020	.008	.0037
900	.....	.....	.....	.....	2.51	.810	.324	.077	.025	.010	.0047
1000	.....	.....	.....	.....	3.06	.980	.403	.095	.031	.013	.006
1200	.....	.....	.....	.....	.....	.....	.581	.137	.045	.018	.008
1500	.....	.....	.....	.....	.....	.....	.....	.213	.060	.028	.013
2000	.....	.....	.....	.....	.....	.....	.....	.390	.125	.050	.023
2500	.....	.....	.....	.....	.....	.....	.....	.....	.196	.078	.034
3000	.....	.....	.....	.....	.....	.....	.....	.....	.270	.115	.052
3500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.152	.073
4000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.200	.094
4500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.253	.117
5000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.145
5500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.175
6000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.208

that the valve will come exactly in the proper position when screwed into the fitting.

Operation and Maintenance

The bearings on the vacuum producer in the basement require lubrication in accordance with the instructions sent with the machine. Also, the wearing surfaces on the various sweeping implements have to be replaced

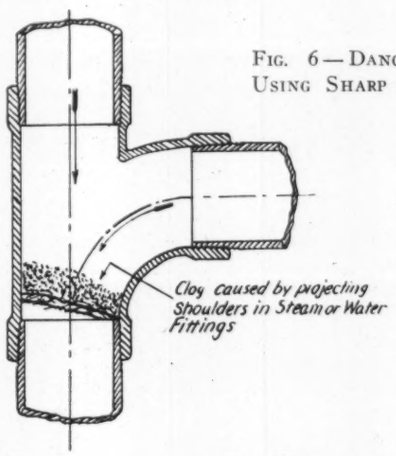


FIG. 6—DANGER OF USING SHARP BENDS

occasionally. Cabinets are usually furnished for the storage of hose and cleaning tools and if these are kept in their proper places when not in use their life is very materially lengthened. Due to the fact that in large systems it often happens that only a portion of the total number of sweepers are in operation at any one time, the velocity of air blowing through the system is materially diminished. Under these conditions it is quite important to flush the system daily so as to eliminate any possibility of dust and litter settling in the horizontal pipe lines. This flushing consists merely of lifting one or two or possibly more valve covers at the remote ends of the system so as to permit a very large volume of air to rush into the system towards the separator tank. This flushing carries away any debris which may be accumulated in the pipes with the result that they are kept clean and the dust and litter is not allowed to accumulate.

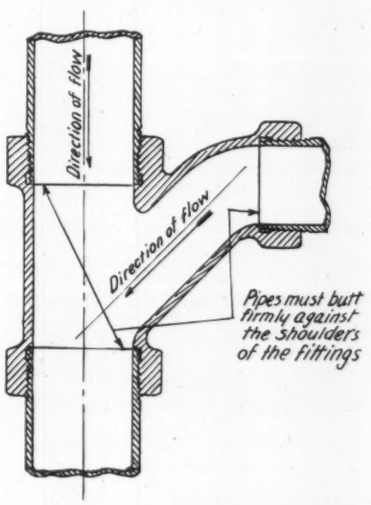


FIG. 7—LONG TURN 90-DEGREE Y BRANCH, TEE PATTERN

It is, of course, essential that the cleaning crew be properly trained when it first begins its work. It is customary practice to assign a foreman to each group of several operators, making him responsible for the work. After the operators are properly broken in originally no further attention or care should be necessary aside from breaking in and training new operators from time to time to replace those who may leave. Accordingly, there is very little that requires the attention of the building manager and operating engineer other than to see that his crew is functioning smoothly and efficiently.

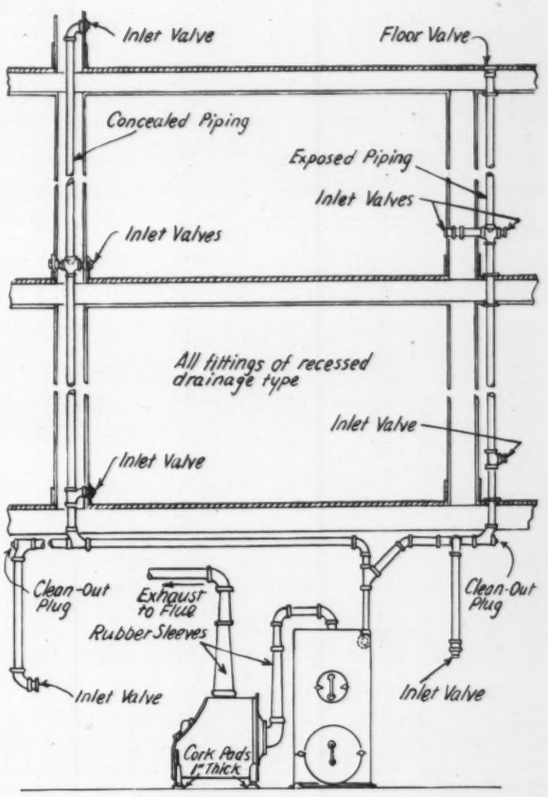


FIG. 8—TYPICAL PIPING ARRANGEMENT FOR VACUUM CLEANING SYSTEM

YOU'LL FIND IN THE NEXT ISSUE

An article on
Heating

The story of how a large armory was heated by gas-fired unit heaters and the way the numerous problems were solved. Told by Louis L. Narowetz, Jr.

An article on
Piping

A discussion of how the flanged joints were designed for 1000-degree Fahrenheit steam at Delray; the second part of Arthur McCutchan's valuable paper, the first installment of which is in this issue.

An article on
Air Conditioning

C. P. Yaglou's report on the influence of atmospheric conditions on health and growth, representing a very complete study of this important subject.

—and many others, of course.

"Open for Discussion"

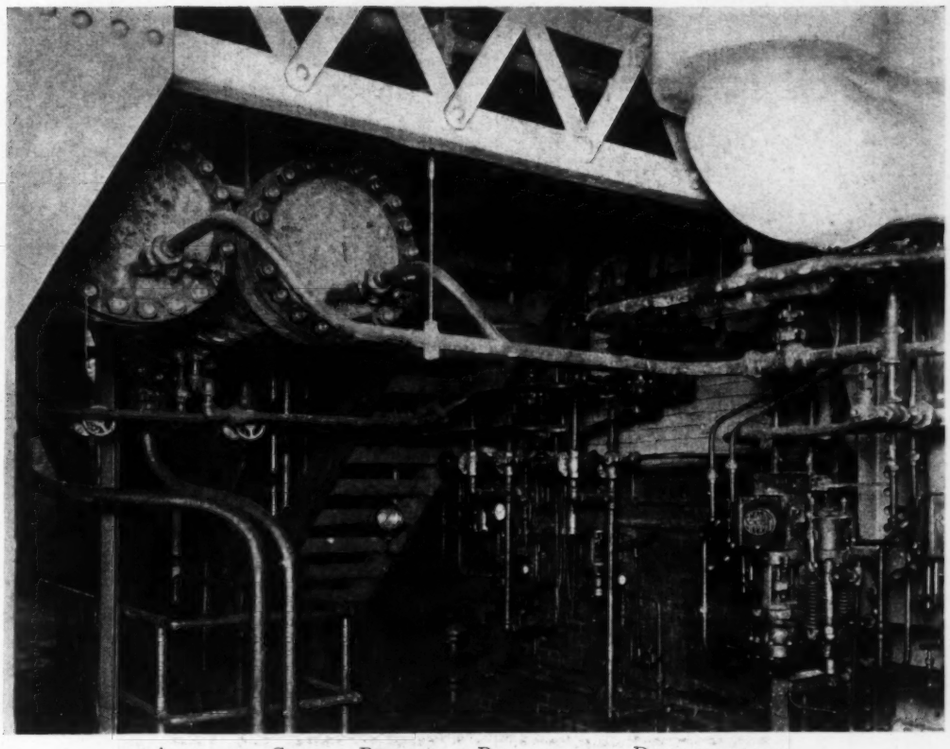
Process Steam and Power

E. H. BARRY'S article "High Pressure Steam and the Modern Industrial Plant" in the July issue is a very complete portrayal of the by-product power possibilities in industrial plants and should be valuable to all plant engineers and executives. It gives a good review of the adoption of higher pressures in central stations and in industrial plants in this country. The author's treatment of the problem of selecting the proper steam conditions and capacities to suit a given problem should be of considerable service to those engaged in this kind of work.

Use of Bleeder Turbines

Turbines exhausting at a back pressure are used in many industrial plants where the electrical lines are tied in with some other source of power which maintains the frequency and takes care of the power demands not supplied as a by-product of the process steam demand. Many other industrial plants depend entirely upon their own plant, avoiding the expense of tie line service.

In making an analysis of this latter type of plant, it is particularly necessary to see that the power generated by the minimmum demand for steam is greater than the



AUTOMATIC CONTROL BOARDS FOR REDUCING AND DESUPERHEATING HIGH PRESSURE STEAM

electrical load that will be required for it will otherwise be necessary to exhaust steam to the atmosphere. In such plants, one of the machines is frequently a bleeder turbine, i.e., a machine that bleeds steam out under pressure control at the desired pressure but passes through further stages of the turbine to the condenser an additional amount of steam sufficient to carry the balance of the electrical load. With this type of machine a kilowatt can still be generated on approximately 5,000 Btu per hour at the coal pile when the process steam demand is greater than the equivalent electrical load but at noon time, holidays, or in case of process shut down, or other unusual conditions, the full load may be carried condensing all the steam and not sending any to the process line or to atmosphere. This is, of course, a valuable type of unit for plants where the process at times requires more steam than the equivalent electrical load and at times less.

Large Savings Made

In many industrial plants, large savings are made possible by turbines exhausting steam to a process, and in other plants by the use of a higher pressure boiler and turbine exhausting steam to the present low pressure header, thereby obtaining the benefits of the greater steam expansion range and at the same time retaining the engines, compressors and other steam equipment throughout the plant which is designed for the lower pressure. Such power may be developed at a cost of 2½ to 6 mills per kilowatt hour, including overhead charges, and in many cases will pay for itself in a period of six months to three years, depending largely upon the load factor of the plant, the greatest savings being made where a high pressure boiler and turbine exhaust steam to a lower pressure boiler plant where no increased supervisory overhead or operating expenses are entailed in the added equipment. Turbine application engineers are often able to determine in a preliminary way whether the potential savings in a given plant will warrant a detailed study being made.

Proper Use and Control of Process Steam

Although frequently the greatest stress in turbine selection is placed on overall economic efficiency, in industrial plants where the heat not converted to work goes to process anyway, equal or greater stress may be placed on the ability to fit a turbine to a given requirement in the proper use and control of process steam.

An Oil Refinery Job

About three years ago, two 1500-kw. turbines were placed in an oil refinery, taking steam at 250 lb. and 125 deg. superheat and bleeding steam at 125 lb.; about ¾ of the steam bled is regained at 5 lb. The turbine regulating system is so arranged that the turbine supplies sufficient steam to take care of the 125-lb. steam demand; under pressure control the turbine governor then admits sufficient 5-lb. steam to carry the balance of the electrical load. The 5-lb. steam in excess of that required to carry the balance of the load by-passes to the condenser which is used as a water heater. If there be insufficient 5-lb. steam to carry the load, the turbine governor by-passes steam directly from the 125-lb. point in the turbine to the 5-lb. steam inlet, as the control is also arranged to maintain the correct pressure in the 5-lb. line.—*S. H. Hemenway,* Industrial Turbine Engineering Division, Westinghouse Electric & Manufacturing Company, Philadelphia, Pa.

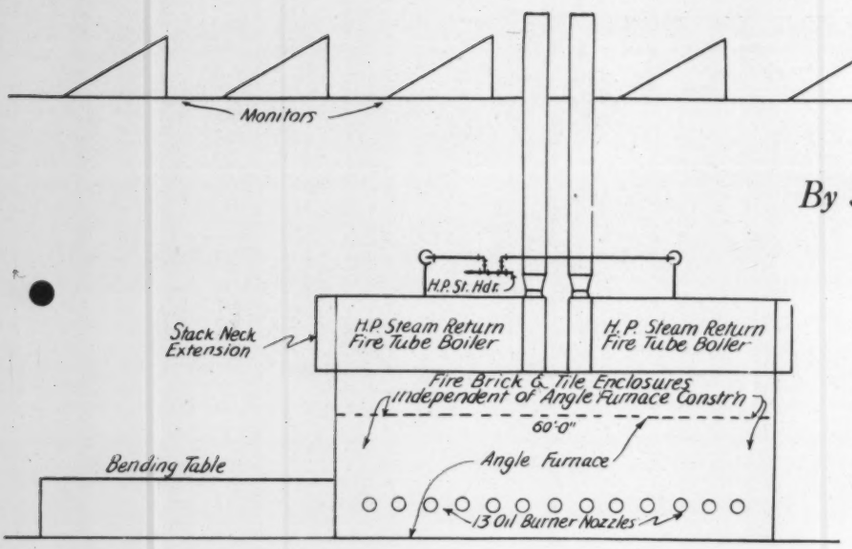


FIG. 1—DIAGRAM OF AN ANGLE FURNACE STEAM INSTALLATION WHICH SAVED SEVERAL THOUSAND DOLLARS IN MATERIALS AND FUEL FOR A SHIPBUILDING COMPANY

Conservation—An Engineering Principle

A Shipbuilding Company Saves Steam—Heat from Diesel Exhaust—An Office Building Oil Burner Control Arrangement

By Josiah C. Moore*

Conservation is a vital part of engineering. It is incumbent upon the consulting engineer to protect not only himself and his client in the design and materials, but to save his client money consistent with good and reliable engineering. This article tells of some interesting savings that were made by one engineer.

FROM the outset of every successful engineering project to its conclusion, some particularly outstanding problem connected with it has influenced the mind of the engineer and given him the courage and ability to master the details and develop a visualization into an actuality. The development of professional endeavor into useful and practical undertakings comes from study and concentration, but the degree of success attained is measured by the versatility of the individual, in the general grasp of the problems presented, its relation to other vital features or contributing elements involved and its value from an investment standpoint over an extended period of years, all of which have a bearing upon the success of the undertaking.

The problem at hand must stand or fall upon its reasonably accurate solution before its release to the public scrutiny, for no successfully developed problem admits of a second attempt or "try." It must be solved upon principles which remove the element of risk to the engineer or his client. The underlying requisites to the successful outcome of an engineering project are education and experience. The latter is perhaps the more important.

Closely coupled to a correct visualization of the problem at hand is the proper presentation. This must follow along the lines of well established methods, often in the form of carefully prepared reports, recommendations and estimates of cost, plans, specifications and details. These documents are all in their order, as a matter of necessity, extremely vital to the success or failure of the project, from both an engineering and a business standpoint, the engineer at all times serving his client as an adviser and his work as rendered, automatically as it were, committing himself to the program and his client to an investment of money.

It is, therefore, all-important that the engineer realize his responsibility, before advising his client and asking him to spend money.

It was with the foregoing thoughts governing the writer's approach and conception of the work to be done which influenced his practice on the following projects. In each instance, mention is made only of certain features or problems, as outstanding in the writer's mind. Underlying each of these problems as solved and the impressions as made upon the writer, is the vital word *"Conservation"*—conservation as applied to engineering.

For instance, during the war the Seattle-North Pacific Shipbuilding Co., Seattle, was engaged in a steel boat building program for the shipping board. Upon the writer's taking charge of the mechanical and electrical departments, orders came fast for many requisites; among

* Consulting equipment engineer, Seattle, Wash.

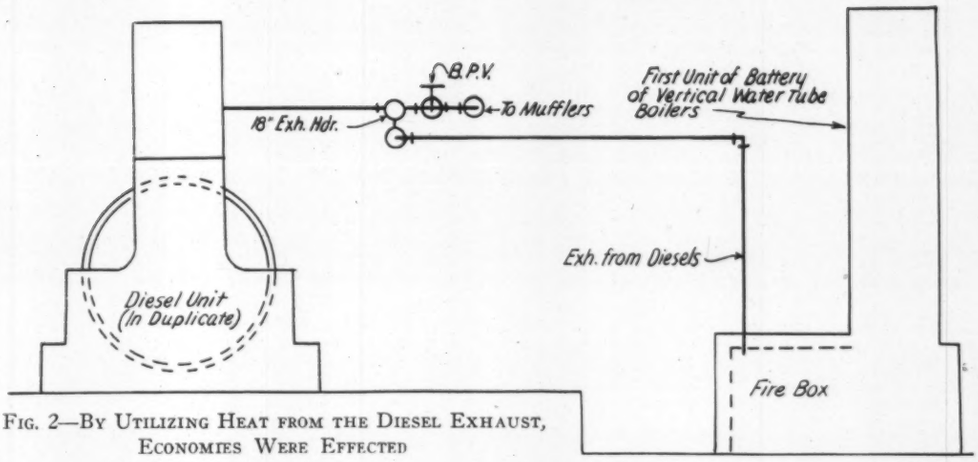


FIG. 2—BY UTILIZING HEAT FROM THE DIESEL EXHAUST, ECONOMIES WERE EFFECTED

others, power and heating plant provisions. All the requests necessitated very prompt action.

Waste Heat from Angle Furnace Utilized

Walking through the plate shed disclosed the fact that there was a very high heat loss from the angle furnace which had a thirteen-oil-burner firing system. All of the waste heat was escaping out of the roof monitors. Here an idea presented itself worthy of careful consideration.

A steel framework was fabricated in the machine shop and erected over the angle furnace in the plate shed. Upon it were placed, end to end, two double stack extension high pressure horizontal tubular boilers with fire brick enclosures and special by-pass systems to regulate the heat. These furnished steam to a vacuum heating system for all of the buildings and high pressure steam for the machine shop. The installation is diagrammed in Fig. 1.

It was found that the results amply justified the scheme, for sufficient high pressure steam was developed from the waste heat to supply the steam hammers in the machine shop and the entire heating system. This represented a saving of approximately $6,000.00 in materials and a saving in fuel of approximately $1,000.00 per month, represented by utilizing a waste product. In addition, the donkey boiler previously installed for supplying steam for the steam hammers was removed and sold.

Heat Recovered from Diesel Exhaust

Several years ago the Kennecott Copper Corporation, Kennecott, Alaska, was seeking relief from very excessive fuel oil outlays for supplying the steam demands of the surface plant and the mines. One of the largest contributing elements of waste was the obsolete condition of the physical systems, principally that of the surface plant, which extended over a large area, comprising heating, hot water, oil tanks and transmission.

Reports and recommendations resulted in a rebuilding and modernizing of both the surface plant system and the mine systems.

At the time of the writer's visit and work in connection with this project, the corporation was installing in its surface plant station two 600 kva. size diesel-driven alternators. These units were planned to exhaust to the mufflers, all heat value thereby being lost.

It was decided to recover useful work from the diesel exhaust waste by means of a shunt of the main line connected into the fire-box of the first vertical water-tube boiler, which was to supply the re-vamped surface plant steam distribution. Fig. 2 illustrates the scheme.

It was most gratifying to find that sufficient steam was available under full diesel load to supply this demand and at off-peak periods a conservative fuel oil burner operation served the purpose.

The saving in fuel oil was very marked. Before the changes were made, there was a charge-back for fuel against the surface plant steam load of a sum in excess of $50,000.00 per annum. Under the new system, a saving of about one-half of this amount was made.

A Synchronized Oil Burner Control

One feature in connection with the draft and the combustion of the oil-fired boiler system in the Northern Life Tower, Seattle, is worthy of note.

A hydraulic regulator was specified and used in connection with the boiler plant, together with a system of bars across the rear and front of the battery, transfer rod, levers and chains, for the purpose of effecting, in single or multiple boiler operation, control of the excess draft by a damper system at the rear of the boiler, synchronized with the control of the oil burner operation at the front of the boiler. This control has resulted in a marked economy in fuel, by closely maintaining a fixed steam pressure irrespective of wind and draft conditions and fluctuations in steam demands of zone and other distribution controls and loads. Fig. 3 shows the method of control.

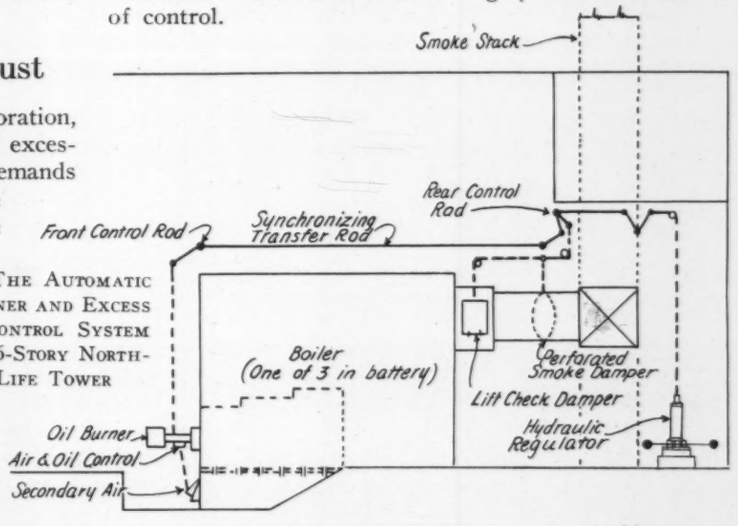
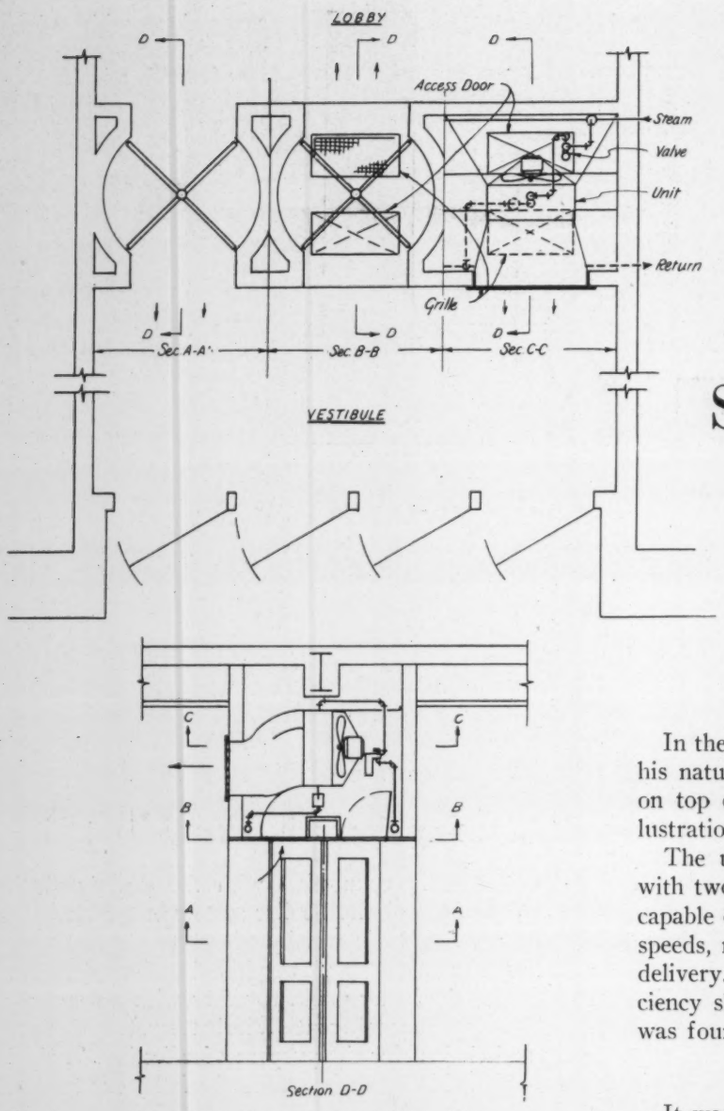


FIG. 3—THE AUTOMATIC OIL BURNER AND EXCESS DRAFT CONTROL SYSTEM IN THE 26-STORY NORTHERN LIFE TOWER



THE INSTALLATION OF UNIT HEATERS SOLVED THIS BUILDING ENTRANCE HEATING PROBLEM

Unit Heaters

Save Floor Space *in* Skyscraper Entrance

By Henry G. Schaefer

OF ALL the mothers of mechanical inventions and expedients, probably none is more prolific than the modern skyscraper. Among the numerous problems encountered by the heating and ventilating engineer, one of the most troublesome is that of adequately heating lobbies, entrance halls, and elevator corridors under the handicap imposed by practically continuously open street doors and the natural drain of the elevator shaft flues.

Generally, just when the poor engineer has emerged from a fog of Btu's with an elaborate scheme of concealed radiation augmented by a recirculating system, along comes one of Mammon's lieutenants to shake him out of his fancied security with a rude and sarcastic reminder that "That floor space is worth $50 per square foot annually and is not available for anything so manifestly foolish as a heating system, even at that figure!"

Unit Heaters Over Revolving Door

In the case of one particular New York skyscraper, all his natural enemies finally treed the harrassed engineer on top of the revolving doors. The accompanying illustration shows what happened.

The unit heaters used in this installation are fitted with two-speed motors (870 and 1,170 r.p.m.), and are capable of delivering 140,000 and 163,000 Btu's at these speeds, respectively, to air entering at 60 F, and at free delivery. The arrangement used decreased their efficiency slightly. Tests were made at both speeds and it was found that the slow speed gave better results.

Control

It was decided to arrange the installation for manual control of the motors, and to use as the controlling factor of the temperature, that of the inlet air. The thermostatic element of the control valve is placed directly in the path of the incoming air. The action of the thermostats is so adjusted as to admit steam when air enters the outside units at 60 F or below, and at 65 F or less in the case of the center unit, which has the function of maintaining that temperature in the elevator lobby.

The action of the other units is to build up, in the vestibule, the "outer guard" of the whole heating system. The capacities of the units, which might appear, casually, to be very liberal, are very close to the total demand when the total estimated maximum daily traffic of 70,000 persons is reached. (There are two entrances.)

The inlet grilles and access doors are made a part of the revolving door construction. The installation, as a whole, represents economy, capacity, flexibility, and simplicity, utilizes room ordinarily wasted, and occupies no floor space. In discussing the installation, the designer refers, with pardonable pride, to his great moral victory—permission to install on the wall of a side corridor the three toggle switches to operate the motors.

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers
JOURNAL SECTION

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Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods

By C. P. Yaglou[1] (*MEMBER*), L. Claribel Benjamin[2] (*NON-MEMBER*), and Sarah P. Choate[3] (*NON-MEMBER*), Boston, Mass.

This paper is the result of research conducted at the Harvard School of Public Health in cooperation with A. S. H. V. E. Research Laboratory

INTENSIVE investigations have until now failed to discover the specific cause of *deadness,* or lack of a stimulating quality, in the air of occupied rooms, even when temperature and humidity are controlled, as contrasted with the air of the open country. Proponents of the *open air treatment* ascribe this quality of freshness to a vital principle which is lost when air is brought indoors, particularly when ventilation is effected by mechanical means.

In recent years, since the carbon dioxide, oxygen, and crowd poison theories have become obsolete, ionization has been suggested as the *air soluble vitamin,* but it has not yet been identified. The virtues of artifically ionized air have been extolled on purely theoretical grounds, with no scientific confirmation whatever.

The object of the present work is to study the problem of ionization in relation to ventilation and health. This paper deals largely with fundamental changes in the ionic condition of the air in occupied rooms ventilated by natural and by mechanical methods, and with the influence of various air conditioning processes upon the ionic content of air. Since the field is comparatively new, a brief discussion on basic principles of atmospheric ionization is also included.

Properties, Formation, and Destruction of Ions

According to Hess,[4] all gases, like electrolytes, contain positively and negatively charged carriers of electricity (*e.g.,* atoms, molecules, or molecular groups), which are called ions. The charge carried by an ion is the elementary charge; namely, 4.77×10^{-10} electrostatic units (E.S.U.). It is believed that this charge is the same in all gases and that it is equal to that carried by the hydrogen ion in the electrolysis of liquids. Because of this charge, ions move under the influence of an electric field and the direction which they take depends upon the sign of their charge.

In general, two classes of ions are recognized; the small or molecular size, and the large or Langevin ions. Owing to this difference in size, the speed which the two classes of ions attain in an electric field differs enormously. The mobility of small ions in ordinary air varies from 1 to 2 cm per second in a field of unit intensity (1 volt per centimeter), and the mobility of large ions varies from 0.01 to 0.0005 cm per second, depending on their size. As a general rule, the mobility of negative ions is somewhat higher than that of positive ions.

Large ions are formed by agglomeration of small positive or negative ions with condensation nuclei, such as dust, fumes, smokes, or drops of water. They are present in great numbers in city air which is polluted with products of combustion from chimneys and from automobile gases. Under such conditions the number of small ions is at a minimum, and it varies inversely with the number of large ions. Between the small and the large ions are the so-called *intermediate* ions, which are formed under certain conditions of humidity. Their mobility varies from about 0.1 to 0.01 cm per second in a unit field. Although this classification is generally adhered to, experiments by the authors and by others[5] show that gaseous ions do not occur predominantly in any well-defined sizes, but that there is a continuous distribution of sizes from the very small to the very large. The frequency of distribution seems to depend largely on weather conditions and on the extent of atmospheric pollution.

In nature, ions are produced by solar radiation, by cosmic rays, and by radioactive changes in the soils of the earth.[6] Strongly ionized gases diffuse through the capillaries of the soil by the aspirating action of the wind and when the barometric pressure falls. It is believed that this soil respiration contributes about 60 per cent of the total ionic content of the air near the surface of the earth.

The state of atmospheric ionization is maintained, more or less, by the simultaneous actions of other natural processes which tend to destroy or to neutralize the ions. The most important of these are (1) recombination of ions of opposite charge to form neutral ions, (2) agglomeration with condensation nuclei to form large ions, and (3) diffusion and adsorption by solid or liquid conductors.

For a more thorough discussion on physical properties and theories of atmospheric ionization, the reader is referred to an excellent treatise by Hess.[6]

Measurement of Ionic Content of Air

The number of positive and negative ions in a unit

[1] Assistant Professor of Industrial Hygiene, Harvard School of Public Health.

[2, 3] Technicians, Department of Industrial Hygiene, Harvard School of Public Health.

[4] See p. 2 of reference 1 at the end of this paper.

[5] See references 2 and 3.

[6] See reference 1.

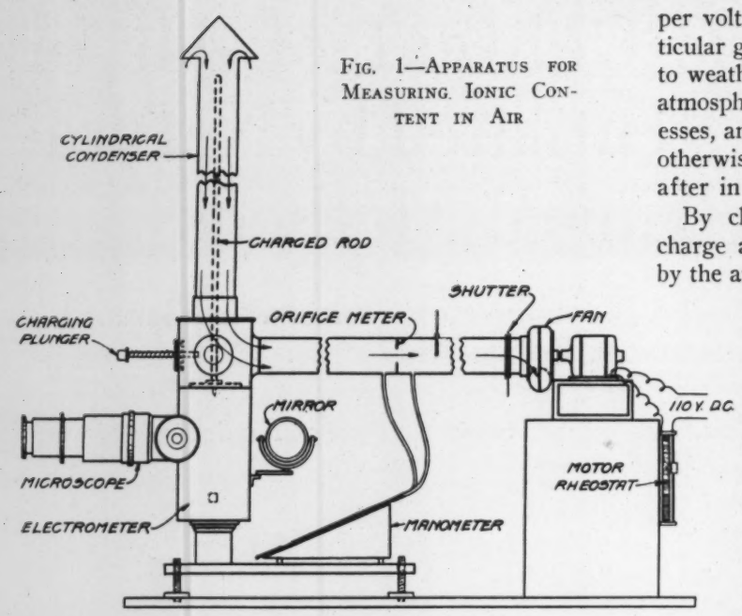

FIG. 1—APPARATUS FOR MEASURING IONIC CONTENT IN AIR

volume of air is usually counted by means of an apparatus devised by Ebert. Fig. 1 shows the modified form of Ebert's apparatus which was used in the studies described in this paper. A stream of air is drawn by fan suction through a cylindrical condenser, the central rod of which is charged to a known potential with a polarity opposite to that of the ions to be counted. The charged rod is well insulated and it is connected to the quartz fibers of an electrometer. All other parts of the apparatus are grounded.

As the air passes down the condenser tube, ions of opposite sign are attracted to the rod, and upon striking it extract a quantity of charge equal to their own. Only those ions will reach the charged rod which have sufficient velocity (mobility) to carry them across the intervening space before they are carried away by the air stream. From the rate of discharge of the electrometer and the rate of air flow—the latter measured by an orifice meter—the number of ions per unit volume of air can be computed.

If $n\pm$ is the number of positive or negative ions in a cubic centimeter of air, w the volume of air passed through the condenser, v_0 the initial charge in volts, v_1 the final charge, corrected for natural leakage if any, e the charge carried by an ion (4.77×10^{-10} E. S. U.), and c the combined electrical capacity of the condenser and the electrometer in E. S. U., the following equation will hold:

$$(n\pm)ewv = \frac{c\,(v_0 - v_1)}{300} \text{ and } n\pm = \frac{c\,(v_0 - v_1)}{300ewv} \qquad (1)$$

The division by 300 gives the loss of potential in E. S. U. The value of $n+$ is obtained by charging the system negatively, and that of $n-$ by a positive charge. Usually it takes five minutes to make an observation. Less time is required when the ionic content is unusually high.

The number and the size of ions caught depend largely upon the design of the instrument, the voltage charge, and the velocity of the air through the condenser. The instrument used by the authors was designed to catch all small ions having mobilities from the maximum possible down to about 0.2 cm. per second per volt per centimeter. The authors adopted this particular group of ions because it was found to be sensitive to weather changes and to changes in the general indoor atmosphere produced by respiratory and metabolic processes, and by indoor activities. Unless specifically stated otherwise, the terms *ions* or *ionic content*, as used hereafter in this paper, refer to this particular group.

By changing the condenser, increasing the voltage charge and reducing the air flow, the instrument used by the authors can be adapted to catch, in addition to the small ions, a large portion of the intermediate and a portion of the heavy ions, as is the case with the usual Ebert's apparatus. From the standpoint of ventilation, however, counts made under these conditions are uncertain, owing to the unknown action of the heavy ions which happen to be near the charged rod of the instrument. When the number of large ions is great, as on dusty and foggy days, those near the charged rod are likely to be caught accidentally, thus vitiating the results. In the apparatus used, the conditions necessary for capture of the small ions are not appreciably exceeded, and therefore the number of large ions caught accidentally is usually very small. When thick dust clouds are raised, however, either by unusually strong winds or by vigorous sweeping and dusting indoors, an appreciable number of large ions are caught accidentally. The only alternative at present is to discard such readings. This is an unfortunate instrumental limitation which the authors hope to correct. In addition to the sources of uncertainty just mentioned, the whole technic of securing accurate results is intricate and laborious, even with the improved apparatus.

Daily measurements (since May, 1930, Sundays excepted) of ionic content of indoor and outdoor air disclose very definite seasonal trends which will be discussed more thoroughly in a subsequent paper. For the purpose of this paper it is sufficient to state that atmospheric ionization undergoes great diurnal and seasonal variations, depending upon local and general meteorologic conditions. It is much higher in summer than in winter, much higher on clear days than on rainy, foggy, or grey days, and, as a general rule, higher in the day time than at night.

The maximum variation in the particular group of small positive or negative ions which were studied (down to a mobility of 0.2 cm. per second) is from about 50 on grey winter days to about 700 on clear days in summer. In contrast with general assumptions, the ionic condition in unoccupied or very lightly occupied rooms does not differ greatly from that out of doors and in cold weather it may even be higher, as will be shown later. However, a few people in a room may alter the situation, even though the amount of air space per occupant is much greater than that which is considered adequate in modern ventilation practice.

Influence of Respiration and Transpiration[7] on Ionic Content in Occupied Rooms

In order to study the influence of room occupancy on ionic content, three different series of experiments were conducted. In the first series, seven persons were seated comfortably in an air tight steel chamber (experimental

[7] Exchange of gases through skin and clothing.

compression chamber, cubical contents 550 cu ft); readings of ionic content were taken both before they entered and continuously while they were in the tank. Smoking was not allowed inside the chamber.

It can be seen from Fig. 2 that both positive and negative ions decreased from an initial concentration of about 250 ions per cubic centimeter to one of about 50, after the people had been in the chamber for 85 minutes. The decrease was very rapid during the first 20 minutes; after that it was gradual. A duplicate experiment, with no one in the chamber and with the ion counter outside, showed no apparent change in the number of ions with time, so that the entire effect must have been due to the occupants and, to some extent, to the de-ionizing action of the ion counter itself. At the beginning, the rate of fall in the number of negative ions was greater than the rate for positive ions; this was probably due to the higher mobility, and hence higher diffusion, of the negative ions to conducting surfaces.

In another series of experiments, ion counts were taken in ordinary classrooms and in reading rooms, under conditions which were fairly representative of those in lecture halls ventilated by the window-gravity methods. Most of these observations were made in a classroom, 30 ft by 33 ft by 10 ft 6 in., having four windows on a northeast wall and three windows on a southeast wall, the window area being about 20 per cent of the floor area. A double door from the room opens directly into the main corridor.

The data shown in Fig. 3 were obtained in this classroom during a luncheon talk in which 34 persons ate luncheon and smoked, while listening to a lecture on current research. No attempt was made to control the general room conditions; the occupants were at liberty to open or close windows and doors, and to come and go as they pleased.

In spite of what might be considered good ventilation (see small temperature rise in Fig. 3), both positive and negative ions fell from about 300 to 65 ions per cubic centimeter in approximately 20 minutes after the people entered, and they remained at this low level until the occupants left the room. It took nearly an hour for the ionic content to resume its initial value after the people departed, all other conditions remaining unaltered. The minimum ionic content in Fig. 3 probably represents the irreducible minimum, due to the influence of radio-active substances in the plaster of the walls and to cosmic rays, which are capable of penetrating thick metal walls and ionizing the air of enclosed spaces[8].

These and many other similar data show that there is a most striking change in the air of occupied rooms as the result of a reduction in ionic content. It makes little difference whether a room is ventilated by natural or by mechanical methods. The problem now is to determine whether such alterations in the electrical quality of air have any significant bearing on health.

The loss of ions in these experiments cannot be accounted for by respiratory processes alone, even if it is assumed that expired air is completely devoid of ions. This is because the volume of air breathed by the occupants (0.28 to 0.42 cfm per person) is very small in proportion to the cubical content of the room (300 cu ft of air space per occupant in the classroom). The de-ionizing action of the ion counter itself is also small (air flow 5 cfm). Transpiratory processes through skin and clothing may account for a considerable portion of the loss, but the problem seems to be complicated by the effects of tobacco smoke, food, and possibly other factors. More data are necessary in order to derive an ionic balance.

[8] See reference 1, p. 117.

Outdoor Air Supply in Relation to Ionic Content

The third series of experiments on the effect of room occupancy on ionic content was carried out in an air-conditioned room, where it was possible to measure the quantity of outdoor air supplied per person per minute in order to maintain normal ionic content. This was done by placing twenty-four persons (men and women seated in armchairs) in the conditioned chamber, and measuring the ionic content at various rates of air change, as shown in Table 1.

Since it was a warm day, the air was cooled to a fairly comfortable temperature by means of pipe coils through which cold brine was circulated. For reasons to be discussed later, the dehumidifier was not used in

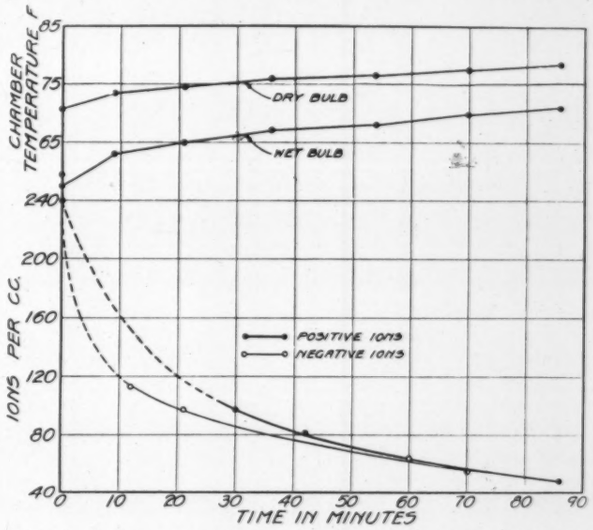

FIG. 2—INFLUENCE OF RESPIRATION AND TRANSPIRATION ON IONIC CONTENT[a]

[a] Seven Persons in a Closed Chamber. Cubical Contents, 550 cu ft. No Ventilation.

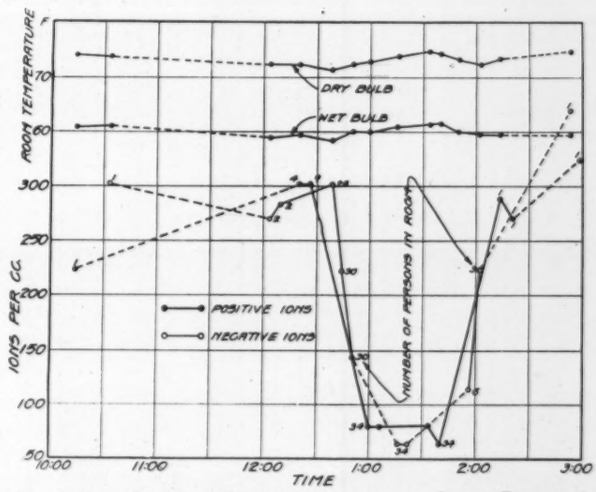

FIG. 3—INFLUENCE OF ROOM OCCUPANCY ON IONIC CONTENT[a]

[a] Room Dimensions 30 ft x 33 ft x 10 ft 6 in. Number of Occupants, Thirty-four. Good Window-Gravity Ventilation.

TABLE 1—OUTDOOR AIR SUPPLY IN RELATION TO IONIC CONTENT

Number of Persons in Room	Ventilation Rate (cfm)		Small Ions per cc			Condition of Air				Remarks
						Supply Registers		Exhaust Registers		
	Total	Per Occupant	Positive ($n+$)	Negative ($n-$)	$\frac{n+}{n-}$	Dry Bulb (Deg. Fahr.)	Wet Bulb (Deg. Fahr.)	Dry Bulb (Deg. Fahr.)	Wet Bulb (Deg. Fahr.)	
2	2050	1025	343	340	1.0	76.0	58.0	76.8	58.4	Initial readings before occupants assembled. All air from out of doors. Cooling by means of central fan system.
24	0	0	95	96	1.0	.....	.....	83.0	71.2	Fans off. Natural air leakage through dampers and crevices. Strong body odors. Minimum ionic content reached in 36 minutes after occupants assembled.
24	2050	85	184	129	1.4	76.0	57.8	80.0	60.5	All air from out of doors. Cooling system in operation. Body odors just perceptible to sense of smell upon entering room.
24	2550	105	207	160	1.3	75.3	56.9	79.2	58.8	All air from out of doors. Cooling system in operation. Body odors not perceptible to sense of smell upon entering room.
24	3450	145	345	303	1.1	77.8	58.0	79.5	59.2	All air from out of doors. Cooling system in operation. Body odors not perceptible to sense of smell on entering room.
24	2050	85	2790	80	35.0	75.0	56.5	78.8	58.5	Artificial ionization. All air from out of doors. Cooling system in operation. Body odors not perceptible to sense of smell on entering room.
1	2050	2050	4810	635	7.6	.....	.....	.....	.....	Artificial ionization. All air from out of doors. Cooling system in operation. After occupants departed.

cooling. When the subjects entered the room, the fans were stopped, and the ionic content was allowed to fall to an irreducible minimum level. The fans were then started again, and were kept running at constant speed until a new equilibrium was established. The procedure was repeated with gradually increased air supply up to the maximum capacity of the fans.

Table 1 gives the experimental conditions and the results secured. A better idea of the relationship between outdoor air supply per occupant and ionic content in the room may be obtained from Fig. 4, which indicates that a ventilation rate as high as 160 cfm per person is barely sufficient to maintain normal ionic content. The ionic content does not appear to be significantly higher with the usual air supply of 30 cfm than with no ventilation. This is consistent with experience in classrooms which were fairly well ventilated by natural methods (see Fig. 3). The minimum ionic content in Fig. 4 appears to be higher than that in Fig. 3, perhaps because no smoking was allowed in the former case when the fans were shut down. When the fans were started again, smoking was quite general among the subjects.

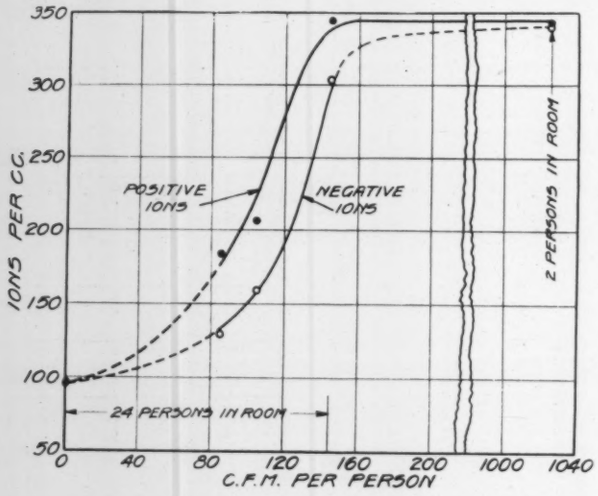

FIG. 4—OUTDOOR AIR SUPPLY IN RELATION TO IONIC CONTENT

Artificial Ionization

From the results of the foregoing experiments, it is quite evident that if the ionic content of air has any beneficial effects on health, some artificial method for ionizing the ventilating current must be employed.

The last two lines in Table 1 give data secured under conditions of artificial ionization. An ionizer was adjusted to produce about 5,000 small positive ions per cubic centimeter of room air (calibration with 2,050 cfm and one person in room) and the least possible number of negative ions, 635 in this case. When there were 24 persons in the room and there was an air supply of 85 cfm per person, the number of small positive ions was increased from 184 to 2,790 per cubic centimeter of air, and the negative ions decreased from 129 to 80. Probably, the difference between the output of the ionizer and the actual ionic content of the air in the room represents the loss by respiration and transpiration.

This and other experiments on artificial ionization indicate that it is fairly practicable to control the ionic content in occupied rooms at any desirable level, up to a maximum of 10,000 ions per cubic centimer, without producing a perceptible quantity of ozone. According to the physiologic experiments of the authors, it is doubtful whether a concentration higher than 2,000 ions would be needed in ventilation work, and experiments are now in progress to determine the threshold value. The maximum ionic content that the authors have recorded in Boston, since May, 1930, has been about 700 ions per cubic centimeter, in clear summer weather. Subjective sensations in artificially ionized air, together with effects on health and physiologic reactions, will be discussed in another paper.

Influence of Air-Conditioning Methods on Ionic Content

There seems to be a growing belief that modern systems of mechanical ventilation deprive the air of its ionic content, by diffusion and adsorption to grounded metal surfaces, and by overheating. Experiments in our psychrometric room failed to confirm this assumption, except when the air was brought in contact with

a fine water spray by passing it through the usual type of dehumidifier.

The influence of various air conditioning processes was studied, both separately and in combination, by taking ion counts in the psychrometric room before and after making similar observations out of doors at the air intake. In this way, diurnal changes in ionic content were accounted for, to some extent; but the compensation was never entirely satisfactory, owing to rapid diurnal changes. Therefore, for the time being, discussion shall be confined to general tendencies only, until a second ion counter is secured for making simultaneous measurements indoors and out of doors.

In the air-conditioning installation used in these experiments, the air supply is taken from an open yard, at a height of about 7 ft above the ground. The air travels a distance of about 100 ft, through a 17 by 22-in. galvanized-iron duct, before it reaches the inlet registers in the psychrometric room. On its way to the room, it passes through seven 90 deg elbows, a dehumidifier (3 ft × 3 ft × 9 ft), six stacks of heaters, or coolers, in series, and two louvre dampers. In the particular experiments which were performed for the purpose of determining the effect of air conditioning processes, there was one person in the room, and the air circulation through the system was 2,050 cfm. This is the minimum air supply with dampers wide open.

With simple ventilation (no air conditioning), the overall loss in ionic content, from the outdoor air intake to breathing zone level in the psychrometric room, varied from 0 per cent to 10 per cent for the positive ions, and from 0 per cent to 30 per cent for the negative ions.

In contrast with the prevailing belief, heating the air by means of heating units used with central fan systems increased both positive and negative ions in all experiments. As a general rule, the increase was sufficient to counteract the loss by diffusion and adsorption to metal surfaces, and in a few instances, the ionic content of the room air was appreciably higher than that out of doors. Cooling, on the other hand, decreased appreciably both positive and negative ions.

These effects of heat and cold are in accord with a well-known connection between temperature and atmospheric ionization.[9] The fact that in cold weather the ionic content in unoccupied heated rooms is often higher than that out of doors, may be explained in part or in whole by this temperature relationship. A good circulation of raw outdoor air through open windows is instrumental in quickly reducing the ionic content in the room to the outdoor level.

The most striking effects of air conditioning were produced when the spray system of the dehumidifier was in operation. Whether the air was washed, humidified, or dehumidified, it was deprived of all small ions having mobilities upward of 0.9 cm per second. Moreover, the spray produced a great number of large negative ions, or condensation nuclei, similar to those produced by hard rain (*Lenard effect*[10]), smoke, or fumes. The higher the water pressure at the sprays, and the dirtier the spray water, the greater was the number of large negative ions. Large positive ions were also present but in comparatively small numbers. Since the instrument was designed to count small ions, it was impossible to catch more than 20,000 intermediate and large ions per cubic centimeter. With all instrumental modifications possible, the computed minimum mobility of the ions caught was 0.03 cm per second, as compared with the minimum known value of 0.0005 cm per second for the largest ions.[11]

Simple recirculation (no air conditioning) gradually reduced both positive and negative ions, and when there were more than three persons in the room the irreducible minimum was substantially the same as that in Fig. 2. In all cases the negative ions were affected more quickly than the positive ions, as is to be expected from their higher mobility.

[9] See reference 1, p. 46.

[10] See reference 1, p. 63.

Summary

A series of experiments was carried out in rooms, both occupied and unoccupied (1) with no ventilation, (2) with window-gravity ventilation, and (3) with mechanical ventilation, in order to determine the extent to which the number of small ions is affected by respiration and transpiration, and by modern air-conditioning methods.

In contrast with the prevailing belief, the ionic content in unoccupied heated rooms did not differ much from that out of doors, and in cold weather it was often higher, owing probably to a temperature effect.

In occupied rooms there was a marked decrease in both positive and negative ions. Immediately after the occupants assembled, the ionic content of the air fell abruptly to a very low value, which was maintained until the occupants left the room. Both positive and negative ions began to rise again as soon as the people departed.

The minimum supply of outdoor air required to maintain normal ionic content in a crowded room was found to be prohibitively high (160 cfm per person). With the usual air supply of 30 cfm per person, the ionic content did not seem to differ greatly from that with no ventilation at all. On the other hand, it was possible by means of artificial ionization to control both the quantity and the quality of ions at any desired concentration up to 10,000 ions per cubic centimeter, with or without ventilation.

Mechanical ventilation reduced the ionic content from 0 per cent to 30 per cent by diffusion and adsorption to metal conductors. Heating the air by means of a central fan system increased the ionic content, and cooling by similar methods decreased it. The usual methods of washing, humidifying, or dehumidifying by means of water sprays, deprived the air of all small ions, and produced a great number of large negative ions, or condensation nuclei, by the well-known *Lenard effect*. Recirculation reduced both positive and negative ions by diffusion and adsorption to metal coductors.

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[11] See reference 1, p. 8.

Surface Coefficients as Affected by Direction of Wind

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This paper is the result of research conducted at the University of Minnesota in cooperation with the A.S.H.V.E. Research Laboratory

IN THE previous work at this laboratory on surface coefficients, the apparatus was arranged so that the direction of the wind flow was parallel to the surface of the material under test. In practice, the wind may blow at any angle to the exposed surface, and the question arises as to the relation of the surface coefficients for different angles of incidence between the wind and the surface.

Apparatus and Test Procedure

In order to determine this relation, test apparatus as shown in Figs. 1, 2 and 3, was set up and a series of tests made with wind velocities varying from 0 to 30 mph and at angles to the test surface varying from 0 deg to 90 deg. The apparatus consisted essentially of a 30-in. air duct 25 ft long, supplied with air from a variable speed fan at velocities varying from 0 to 30 mph. The air duct was provided with a Pitot tube and draft gage at a point 75 in. from the outlet end for measuring the air velocities. The test surfaces were 15 in. square and were placed with the center line 12 in. in front of the outlet end of the duct. Fig. 1 shows the fan, the 30-in. duct, and the Pitot tube arrangement with gages. Fig. 2 shows the open end of the 30-in. duct with the test specimen in place, together with the rheostat for controlling the temperatures of the test surface.

Fig. 3 shows a plan view of the outlet end of the air duct, together with a partial sectional view of the test surface and heat meter. The test surface proper was 15 in. square, with a 12-in. wing or extension wall on the leading side in order to direct the wind over the surface and to prevent the disturbing eddy currents from the leading edge of the test plate.

The test surface was placed against a meter which was 12 in. square and which was supplied by heat with an electrically heated plate and rheostat. The meter was similar to the Nicholl's heat meter and was the same one used in previous tests reported in the Journal of the Society (*Heating, Piping and Air Conditioning*) June, 1930. As shown in Figs. 2 and 3, the meter was placed on a pedestal and so arranged that it could be rotated on its vertical axis at any angle to the direction of the wind. The wind velocity from the duct could be varied from 0 to 30 mph; therefore, a wide range of test conditions was possible.

The differential temperatures of the heat meter,

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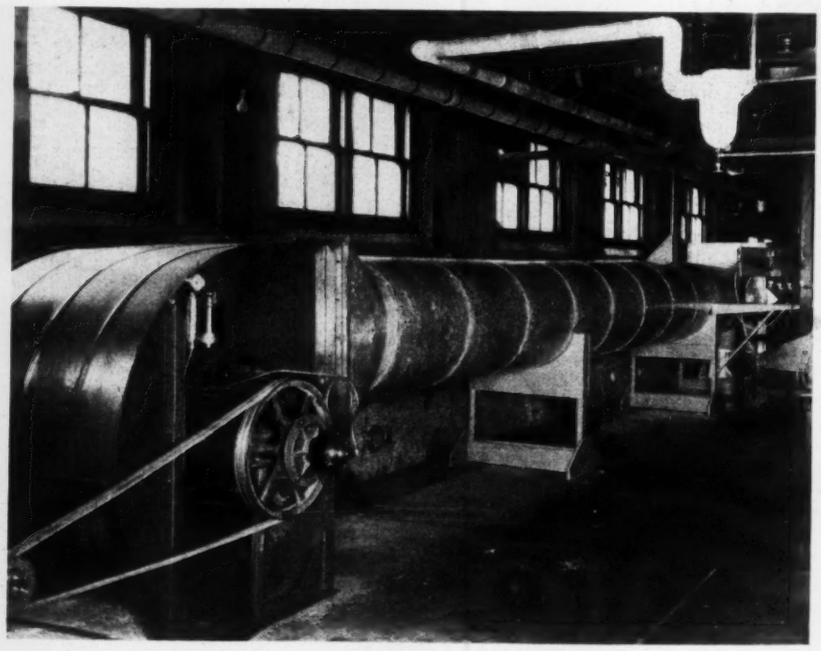

FIG. 1 — VIEW OF FAN, 30-IN. DUCT, AND PITOT TUBE STATION

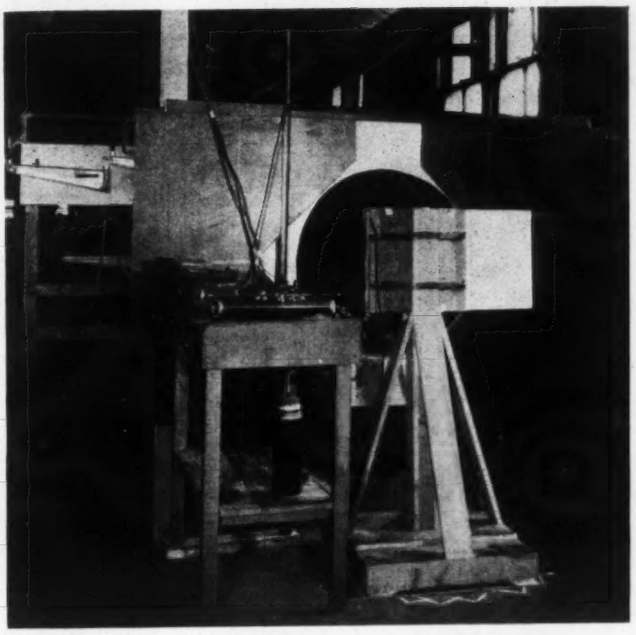

FIG. 2—VIEW OF OUTLET OF 30-IN. DUCT SHOWING TEST SURFACE AND APPARATUS

TABLE 1—AIR VELOCITY AND STATIC PRESSURE AT SURFACE FOR VARIOUS ANGLES OF INCIDENCE OF WIND TO SURFACE

ANGLE OF INCIDENCE (DEGREES)	WIND VELOCITY (MILES PER HOUR)	VELOCITY PRESSURE IN DUCT (IN. OF WATER)	VELOCITY PRESSURE PARALLEL TO SURFACE	WIND VELOCITY PARALLEL TO SURFACE	STATIC PRESSURE AT SURFACE (IN. OF WATER)
0	10	0.0465	0.0420	9.50	0.000
	15	0.1045	0.072	12.46	0.004
	20	0.185	0.170	19.12	0.010
	25	0.290	0.310	25.85	0.019
15	10	0.0465	0.052	10.59	0.004
	15	0.1045	0.089	13.83	0.007
	20	0.185	0.192	20.35	0.011
	25	0.290	0.320	26.25	0.016
30	10	0.0465	0.055	10.85	0.016
	15	0.1045	0.1045	15	0.032
	20	0.185	0.190	20.22	0.054
	25	0.290	0.308	25.75	0.085
45	10	0.0465	0.0465	10	0.030
	15	0.1045	0.1045	15	0.068
	20	0.185	0.185	20	0.120
	25	0.290	0.306	25.67	0.179
60	10	0.0465	0.032	8.3	0.042
	15	0.1045	0.064	11.74	0.096
	20	0.185	0.112	15.5	0.182
	25	0.290	0.192	20.3	0.267
75	10	0.0465	0.013	5.29	0.050
	15	0.1045	0.024	7.19	0.115
	20	0.185	0.042	9.51	0.195
	25	0.290	0.076	12.8	0.309
90	10	0.0465	0.002	2.07	0.053
	15	0.1045	0.004	2.94	0.125
	20	0.185	0.009	4.4	0.230
	25	0.290	0.020	6.57	0.360

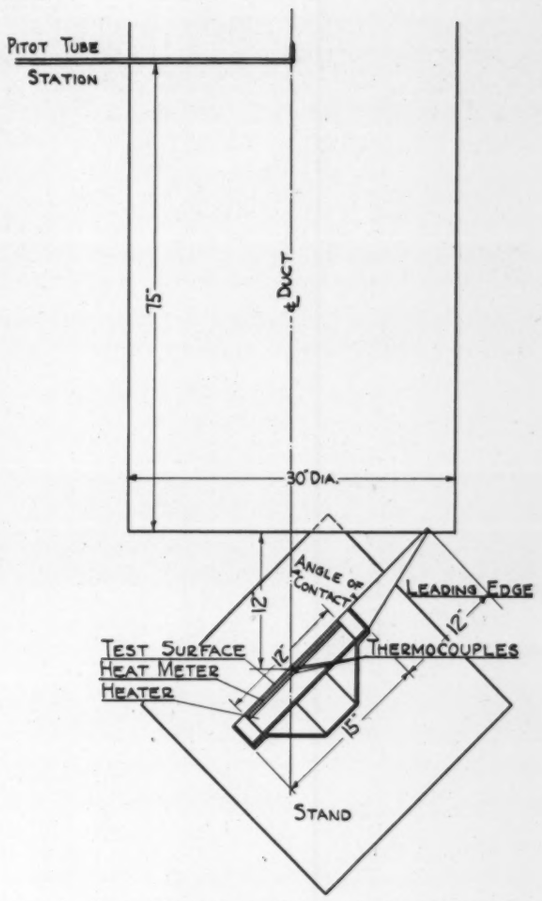

FIG. 3—PLAN VIEW SHOWING RELATION OF TEST SECTION TO OUTLET END OF AIR DUCT

the surface temperatures of the test specimen, and the air temperatures were taken with copper constantan thermocouples and a potentiometer. The surface temperatures were taken by 28-gage copper constantan thermocouples flattened out and cemented to the surface of the test specimen with thin vellum paper. The air temperatures were taken by a thermocouple placed 1½ in. in front of the test surface. The air velocities in the duct were taken with a Pitot tube 75 in. from the outlet end, and again, by the Pitot tube placed close to the test surface to determine the velocities parallel to the test surface, and also, the static pressures of the air at the surface. In making the tests, plate glass and smooth pine surfaces were used, and tests were made at angles varying from 0 to 90 deg through 15 deg increments. For each angle, the velocity in the air duct was varied from 0 to nearly 30 mph.

In the set-up as made, it was impossible to vary the air temperature through any wide range. Since the object of the test was to find the relation between the coefficients at different wind velocities, a mean temperature was selected which was within the range of the apparatus, and this was approximated throughout all the tests. The temperature of 83 F was maintained throughout most of the tests, although in some cases there was a variation in either direction as high as 5 deg in the mean temperature. This variation was not sufficient to make any practical difference in the results. The mean temperature was taken as the aver-

age between the test surface temperature and the surrounding air temperature, it being assumed that the surrounding objects were of the same temperature as the air.

Results of Tests

The results of the tests are shown graphically in the curves of Figs. 4 and 5. By comparing these curves with those obtained in previous tests for parallel flow, it was found that the coefficients at zero velocity are substantially the same. At 15 mph, the coefficients for both pine and glass are slightly greater, and at 30 mph the coefficient is approximately 10 per cent greater for glass and 17 per cent greater for pine, the essential difference in the two sets of curves being that, in the previous set of tests, the coefficients do not follow a straight line, but as the wind velocity increases, the rate of increase for the coefficients is diminished, whereas in the present tests, the curves are substantially straight lines.

As the angle at which the wind struck the test surface was changed from zero, the coefficient was slightly reduced. For wind velocities up to 15 mph the coefficients were substantially the same for angles of 15 deg to 90 deg, all being less than for parallel flow. Above a 15-mph velocity, the coefficients were reduced as the angle was increased. On the whole, the reduction in the numerical value of the coefficient was not as much as was anticipated, and for practical purposes, the coefficients as obtained for parallel flow would be satisfactory.

In order to show the action of the air on the surface for the different angles of wind to surface, readings were taken to determine the velocity and static pressure of the air near the test surface for different angles and for different velocities in the air duct. The results of these readings are shown in Table 1. From these data, it will be observed that, as the angle of incidence is increased, the static pressure is gradually increased at the surface until at 60 deg the static pressure practically equals the velocity pressure in the main duct. At 75 deg and 90 deg, it slightly passes the velocity pressure. Further, as the angle of incidence is increased, the velocity pressure on the surface, and therefore the surface velocity, are substantially the same as the velocity of the air in the duct until an angle of 45 deg is reached, after which the surface velocity is gradually reduced until it reaches a minimum at 90 deg.

Since the surface velocity is very greatly reduced for the high angles of incidence, it might be assumed that the surface coefficients for these conditions should also be greatly reduced. The fact that they are not is probably due to the corresponding increase in air pressure at the surface, which makes the contact between the air and surface more effective in removing the surface heat.

As a further study to determine the action of the air on the test surface for the various angles of incidence between the air and the surface, sheet metal plates, 12 in. x 18 in. in area, were placed perpendicular to the test wall and in the plane of the air flow. In this position, the plates did not disturb the air flow but merely separated it as it approached the wall. A light coating of lamp black and kerosene was placed on the surface of the metal sheets and the air was blown on to the test surface for a sufficient length of time to impress or mark the direction lines of the air on the surface of the plates. The plates as obtained in these tests are shown in Figs. 6 to 12, inclusive. These results show very clearly the change in surface velocity conditions which take place after passing the 45 deg angle. At

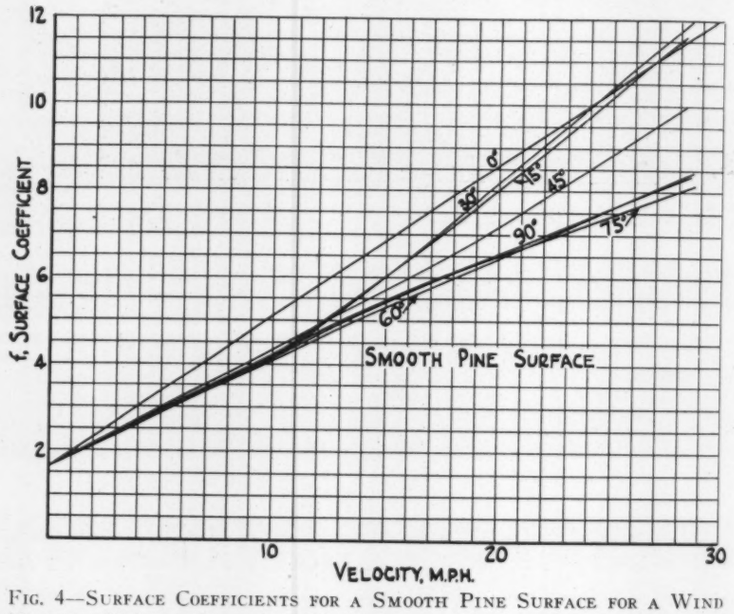

Fig. 4—Surface Coefficients for a Smooth Pine Surface for a Wind Velocity Varying from 0 to 30 Mph and a Wind Direction Varying from 0 Deg to 90 Deg to Test Surface

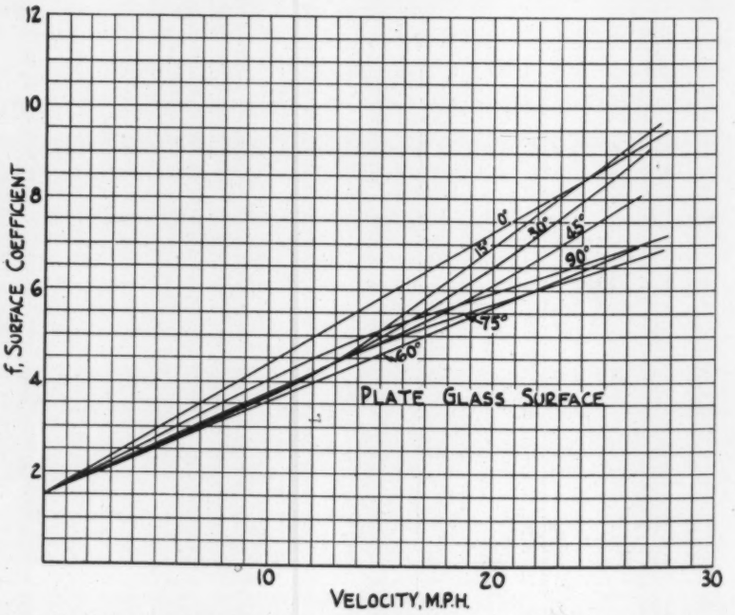

Fig. 5—Surface Coefficients for a Plate Glass Surface for Wind Velocities Varying from 0 to 30 Mph and Wind Direction Varying from 0 Deg to 90 Deg to Test Surface

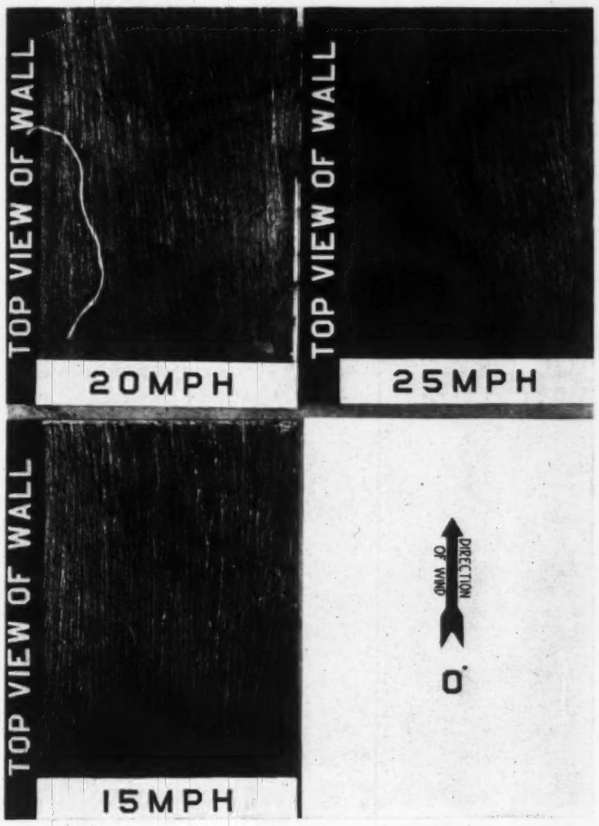

FIG. 6—LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM TEST SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 0 DEG

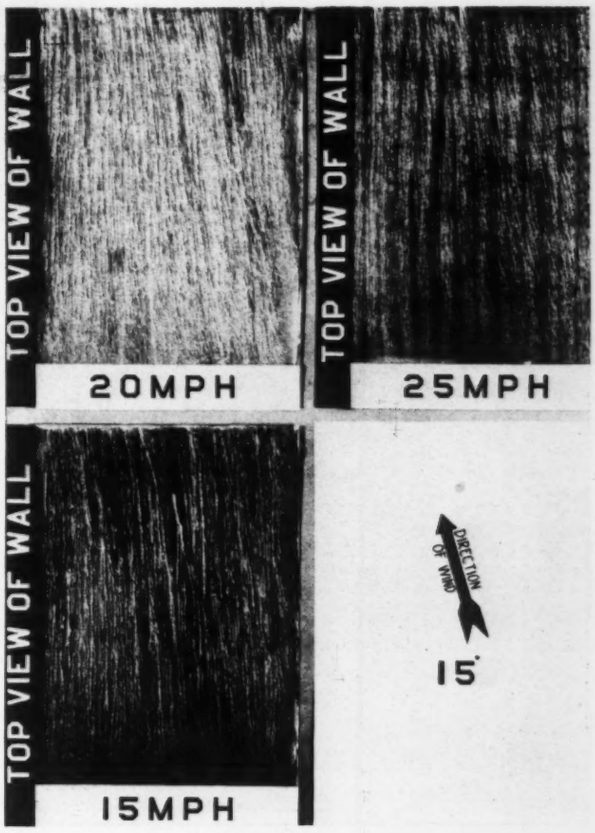

FIG. 7—LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 15 DEG

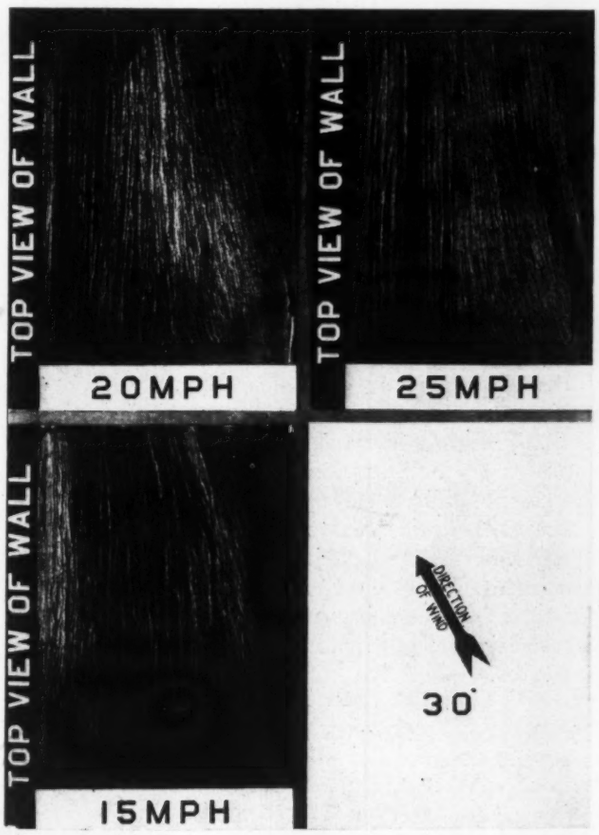

FIG. 8—LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM TEST SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 30 DEG

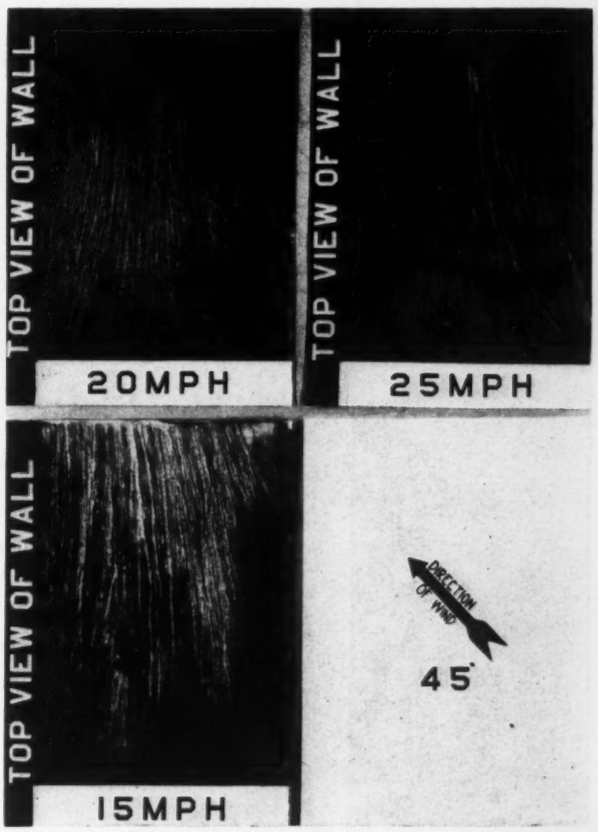

FIG. 9—LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM TEST SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 45 DEG

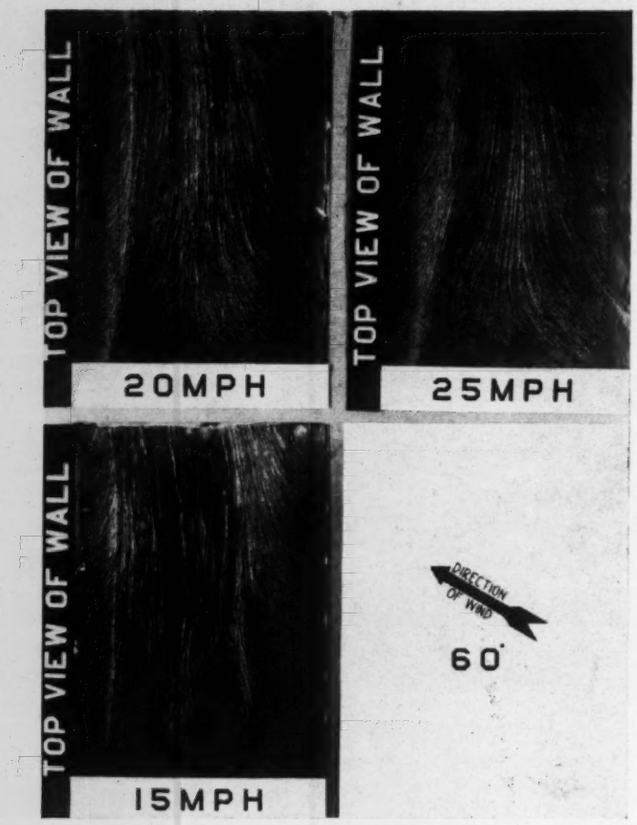

FIG. 10—LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM TEST SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 60 DEG

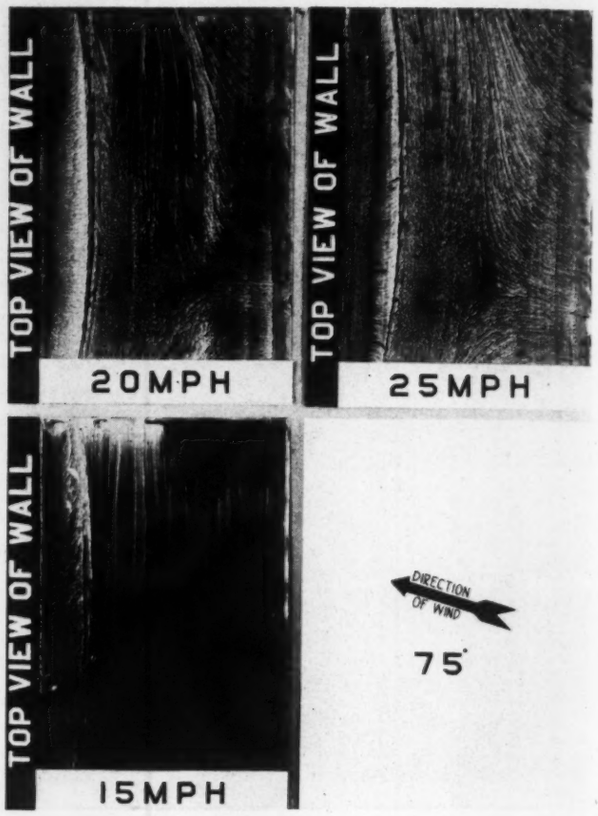

FIG. 11—LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 75 DEG

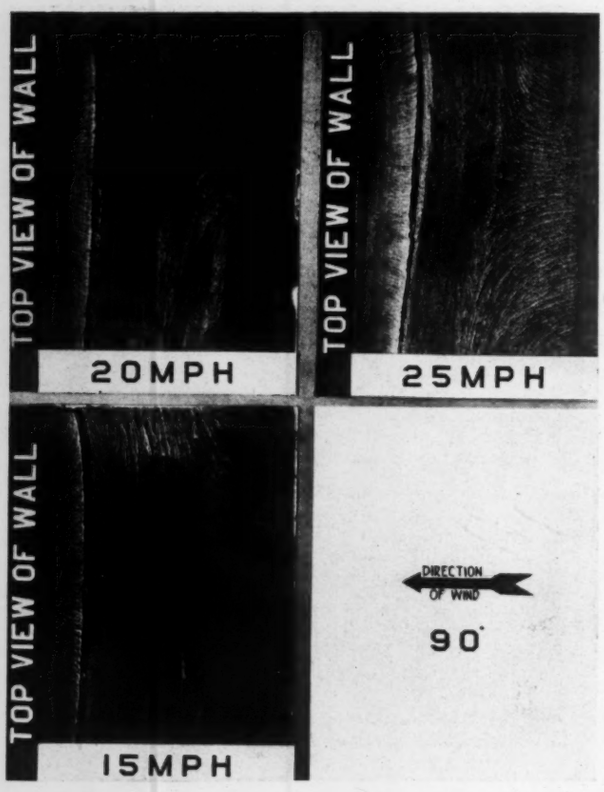

FIG. 12—LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM TEST SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 90 DEG

angles of 60 deg, 75 deg and 90 deg, it is very evident that the surface velocity is retarded but that the surface pressure is building up as was indicated by the pressure gage.

Summary and Conclusions

The results of these tests are significant, *first,* as an indication of what takes place near the surface of a wall as the angle of incidence between the surface and the wind is changed, and *second,* in that, even though there is a changed condition at the surface, the combined effect of this changed condition is such as to make the surface coefficient substantially the same as that obtained for parallel air flow. For all practical purposes, it would appear that the surface coefficients as obtained for air flow parallel to the surface might well be used without any correction for the angle of the wind. Such a correction would obviously be difficult to make, and due to the many other uncertainties surrounding this part of the problem, it would not seem justifiable to go to any such refinement. Tests were made for only one mean temperature, but there is no reason to believe that similar relations would not hold for other mean temperatures.

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Problems and Methods of Heating and Ventilating on Shipboard

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THERE are two factors that present difficulty at the outset in the design of heating and ventilating systems for use aboard a ship, namely, *weight and space* or *lack of space.*

Weight reduction is especially important on modern naval vessels due to limitation agreements with foreign powers. This means that weight restriction must be carefully considered on all component parts of the vessel, consistent with proper strength and rigidity, in order that the required equipment may be placed on board. As to space, it need only be pointed out that a ship may be likened to a floating city, self-sustaining and modern in every respect, from the power plant to its fine table service, and all within the confines of a few thousand square feet.

The location of rooms to be heated and ventilated is another factor that requires careful consideration. Ships are unavoidably so constructed as to produce what are called *inside and outside* rooms, much the same as encountered ashore in buildings. Outside rooms are those bounded by the shell and weather decks, and inside rooms are those separated by passageways and non-weather decks.

Many of the outside rooms above the water line are subject to sea and spray, in other words, while at sea the shell bounding these rooms would have a temperature nearly equal to the wet-bulb reading. The shell is of course insulated, but nevertheless there is considerable heat loss. On the other hand, due to the large amount of heat generated by the ship's boilers (lagging notwithstanding) there is a tendency to overheat the inside rooms. Thus there may be rooms of opposite conditions separated simply by a passageway. This brings up the problem of heat control which will be discussed later.

Generally speaking, naval ships may operate in any navigable sea or river the world over. Therefore, it will be readily seen that a wide range of climatic conditions is encountered. This is also true of certain classes of merchant ships. The first question to be answered in the latter class then is, In what trade does this particular ship operate? Is it a stated trade, say between Liverpool and New York, or say Hamburg and South American ports, or again, Philadelphia and San Francisco? From this it will be readily seen that a ship may run the gauntlet of varying climates in a comparatively short space of time or if in the North Atlantic trade or similar, the changes will not be so wide in range and will be more or less gradual. Consequently, the heating and ventilating arrangements aboard ship may be required to meet one particular climate or a wide range of conditions.

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Naval Ships

The heating, ventilation and air conditioning requirements for naval vessels are carefully worked out by the Navy Department, Bureau of Construction and Repair at Washington, D. C., and it becomes the task of the shipbuilder to design the equipment to meet these requirements.

It may be well at this time to subdivide the ship into the various classes of rooms or spaces, the names of which will convey at once an idea of their nature and what would be expected to maintain bodily comfort. These are officer's quarters, officer's mess, crew's berthing and messing spaces, sick bay, superstructure, consisting of pilot house, chart room, conning tower, etc., general work shops, machinery spaces, boiler rooms, central station, interior communication and plotting rooms, magazines and store rooms, galleys, bakeries, etc.

Heating of Naval Vessels

Heating of modern naval vessels is almost entirely done with steam, the greater percentage of this being done by the indirect method. This steam is generated in the main boilers and is reduced to 35 lb for the heating system in general. In the case of the U. S. S. Indianapolis now building, where superheat is used for propulsion, a desuperheater is used for the purpose of producing saturated steam for the heating and galley equipment, this latter equipment not being designed to stand the superheat. Living quarters and working spaces other than propelling machinery spaces are heated. Magazines and some store rooms are not heated.

Heating is accomplished principally by the use of unit indirect heaters which consist of a double chambered steam header into the top and inner diaphragms of which are fitted a series of tubes, one inside the other, the outer tube being of the fin type blanked at the top end. The steam is supplied from one chamber up the small inner tube, then down the outer fin tube into the exhaust chamber. These steam tubes are suitably encased in sheet steel to form the air passage. The heater box thus formed is lagged with asbestos. These heaters have a cold air by-pass controlled by a manually operated mixing damper.

Fans are connected directly to the heater, and the duct work is led from the discharge to the various spaces in the usual manner. In order to conserve space the fans are mounted on the heaters. This, however, is at present confined to units of 4000 cfm capacity and less.

Humidifying is accomplished by introducing steam at the heater discharge by means of a spray head. Considerable difficulty has been experienced previously from the high rate of condensation at this point. This, how-

ever, has been partially overcome by changing the design of the spray to present a minimum amount of surface to the air flow. The humidifying is controlled manually. Hygrostats are used on some ships but generally are not satisfactory for this service.

The aforementioned heaters will be termed *unit heaters* for convenience. The steam supply to these heaters is controlled by a thermostat so located as to meet the average requirements, and herein lies trouble because it is not always possible to provide a separate system of temperature regulation for the inside and outside rooms. This, of course, means too much or not enough heat, which means that individual control of heat is necessary for complete satisfaction. In the latest naval ships this difficulty is being overcome by the installation of individual heaters in each room, which is, of course, the only practical solution, notwithstanding the added cost. Enough air is supplied for ventilation only, the air being blown through or by-passed around the heating unit as required. Isolated rooms requiring heat are supplied with direct radiation. Where individual heaters are not used, such as in officers' outside staterooms, the warm air supplied for heat and ventilation is supplemented by small electric heaters.

Naval Ventilation

The lack of space presents difficulties which produce conditions not generally encountered ashore. This problem is overcome by increasing velocities which may be as high as 4000 fpm, which of course is considerably higher than would be tolerated ashore for purely ventilation purposes. The terminal velocities are of course never as high as this, but range from 1000 to 1500 fpm, depending on the class of room, as compared to the low terminal velocities of shore installations of say 120 to 300 fpm.

The supply to the fans cannot be taken from the weather at random because of the sweep of the guns. It is also desirable to have as much clear weather deck space as possible for other reasons such as drills, etc. Then again, protection of the intakes from damage by green seas and their possible damage by the handling of equipment such as small boats, stores, etc., all are to be considered. These items, together with the use of mechanical ventilation, are reducing the number of cowls which heretofore were generously sprinkled about, and in their place, wherever possible, the intake trunks are built in as part of the superstructure.

The cowl from a purely ventilation standpoint cannot be surpassed for efficiency, but because of the foregoing objections, together with the fact that many of the cowls have to be unshipped when going into action, makes it more or less imperative to sacrifice efficiency for ease of handling and liability of damage.

It is not possible completely to eliminate intakes from the open deck. However, where conditions make this imperative, a type of inlet known as *rain and spray proof mushroom hood* is used, and while these inlets do what the name implies, they will not prevent a sea from entering. To overcome this a watertight cover is fitted in the deck; this would suggest that the system would cease to function, but another cover is fitted in the intake to the fan below the deck, which is opened and the system becomes a recirculating one.

Mushroom vent hoods are low in stature by reason of the need to clear the line of fire of guns, and when a ship has a moderately low freeboard, the deck is awash even in a fair seaway. For this reason these hoods are not entirely satisfactory, but under the existing conditions must suffice until something better is developed. It need hardly be mentioned that the greatest care is exercised to preserve the water and gas-tight integrity of the vessel. The consequences of neglect of these items are too apparent even to discuss.

In connection with water and gas-tightness, it may be well to note that a ship is subdivided into many water-tight compartments, each designed to withstand a given water pressure. Therefore, when a vent duct terminates in one of these spaces a water-tight cover is fitted, usually where it enters, in order that a minimum amount of the heavier water-tight duct be used. In other words, every water-tight compartment must be absolutely independent of every other.

In the case of magazines which are equipped with flooding systems, the water-tight covers in the exhaust ducts are fitted with swinging check valves so that an air escape is provided when the magazine is flooded. These exhaust ducts are made water-tight to a level which is approximately that of the water line. Above this point they are not water-tight in order that undue stress will not be imposed upon the structure surrounding that particular magazine, which might result from the head of an accumulation of water in the water-tight duct.

When it is realized that terrific stress is placed on almost all parts of a ship, especially in heavy weather, and that the displacement tonnage is limited, it will be apparent that ducts which pierce the structure must be kept to a minimum size in order not to seriously affect the strength. This, together with the fact that a certain amount of head room has to be maintained, is the reason for the high velocities used and their consequent relatively high static pressures.

Ballistic resistance is preserved by the limiting of unprotected openings to 5 in. In the case of ventilation this is accomplished by the use of a Venturi tube with a 5-in. diameter throat. These tubes are used in mechanical supply systems in pipes from 5½ in. to 8½ in. and in natural ventilation pipes from 5½ in. to 10½ in. For pipes exceeding these size so-called armor gratings are used. Armor gratings are of various types. The cast variety is of a very high ade manganese steel, the divisions or webs of which streamlined to reduce resistance. The second type is b t up of S. T. S. bars placed on edge, the maximum sp ing of the bars being about 3¼ in. The third type co sists of S. T. S. plates 3 or 6 in. thick, depending on the thickness of the deck in which they are used. These are perforated with 5-in. diameter holes, the edges of which are chamfered.

Fire protection of magazines and other spaces requiring it is secured by the use of close mesh wire screens fitted in the duct to prevent the entrance of any burning or lighted substance.

It need hardly be pointed out that Venturi tubes, armor gratings and the close mesh screens produce resistance that must be carefully accounted for. The Bureau of Construction and Repair has made thorough

tests of these devices and their resistances can be accurately calculated.

Tests

Rigid tests are made of all ventilation systems on naval vessels, the electrical input method being used for the volumetric test. The anemometer is used at the inspector's discretion on terminals of which he may be doubtful. Tests are prescribed by the Navy Department and are thoroughly carried out under the observance of a representative of the Naval Constructor.

The fan manufacturer is required to make a so-called shop test of one or more fans of each size and type. From this test the fan characteristic curves are plotted which form the basis of the ship test and are used with tachometer, volt and anemometer readings for estimating the volume of air delivered.

In the writer's opinion, the two most outstanding needs for naval ventilation are (1) a propeller type fan that will produce static pressures of 2 in. or over without intolerable noise, and (2) a quick acting water-tight closing device that will abolish the present portable and sliding sections of ducts that are generally required by the present arrangement. The first may be of interest to the heating and ventilating engineer, whereas the second is purely a shipbuilder's problem.

Merchant Ships

In the great trans-Atlantic liners of today the greatest care is exercised in the proper conditioning of air for both the passengers and crew. Notable examples are the Bremen and Europa of the North German Lloyd Line. On the Bremen, as an illustration of the proportion of the installation, 882,833 cfm of air are handled by a total of 115 fans, 58 of which are supply, the remaining 57 being exhaust, all for the sole purpose of heating and ventilation.

The ships referred to specifically and those of the same class are, of course, the extreme. However, very little is left to be desired in any respect. The comfort of the passenger is paramount, and to this end are all the features of the vessel designed and built.

The ventilation requirements for living spaces are outlined by the *U. S. Steamboat Inspection Service,* for the United States and by the *British Board of Trade* for Great Britain. The laws set forth by these organizations are not complicated; rather they are concise and are mainly concerned with an adequate air change within specified time limits for the various classes of passengers.

The classification societies such as the *American Bureau of Shipping* and *Lloyds* are concerned only with the scantlings of ventilators, that is to say, they are interested in whether the vents which affect the water-tight integrity are strong enough. They specify the height and thickness of the water-tight portion above the deck and the connections thereto, but are not otherwise interested.

Freighters, tankers, fruit ships, ships carrying live stock, etc., all present their own particular problems. On these classes of ships very little mechanical ventilation is used for living quarters. Fresh air is usually supplied through the ports or by small natural ventilators. Heating is accomplished by means of steam radiators or coils.

The cargos of these ships require a circulation of air in most cases, and this is generally done by natural ventilation, with the exception of the fruit ships. The old fashioned cowl is the type most generally used for this purpose. Each hold has at least two, one fitted on each end. These cowls can be turned and secured in any direction in a horizontal plane. There have been many attempts made to improve this type of inlet with little or no success.

One of the finest examples of what is being done by temperature control is that of the fruit ships. A cargo of bananas loaded at Kingston, Jamaica, can be discharged at German ports, thence overland into Asia, without the loss of a single piece of fruit due to over-ripeness. Likewise a cargo of the same fruit can be made ready for immediate sale upon arrival at American or European ports.

Fruit ships are generally fitted with a bank of brine coils through which air is forced by large blowers. The shell and decks are amply insulated and the fruit is kept away from the shell by closely spaced wooden battens. The spaces thus formed constitute the air passages.

On passenger ships care is taken to avoid objectionable appearance of equipment. In the case of ventilation it is generally possible to conceal the ducts, and the terminals are designed to meet the architectural scheme.

On men of war it is not possible to conceal the various systems and indeed it may not be desirable if it were possible. These ships are built for a stern purpose, and ready access to all parts is necessary. Nevertheless the piping and duct work is laid out with as much symmetry as possible.

One other item of ventilation that cannot be passed without mention is the windsail. This is an indispensable item of equipment on all merchant ships, especially those carrying cattle. It is in reality an air scoop made of canvas, and is used to supplement the fixed cowls in fair weather.

In conclusion, attention is called to the fact that while heating and ventilating methods aboard ships have been improved considerably during the last two decades, these improvements have not been basically radical in nature and have been rather slow in process. This may be due to the fact that naval architects have their hands full with the general design or are waiting for the manufacturers of this type of equipment. In any case, smaller equipment with a minimum expenditure of horsepower is needed.

Radiant Heating

Most heating systems in use in the United States at the present time involve the transfer of heat by radiation and convection. The *Panel System,* which has been in vogue in England for many years, and which has recently been introduced in the United States, pertains almost entirely to radiant heating.

With this system heat is absorbed by the body and clothes at such a rate as to maintain comfort, even though the surrounding air is comparatively cool. The lower air temperature causes the indoor air to be intrinsically more humid in the winter and more pleasant for the skin and mucous membranes. The heat losses from the building are likewise reduced.

U. S. Naval Requirements for Ship Heating and Ventilation

By E. L. Gayhart[1] (*NON-MEMBER*)
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THE subject of the heating and ventilation of ships falls under two heads: (1) the statement of the requirements imposed by the design Bureau of the Navy Department upon the ship contractor and (2) the methods adopted and problems encountered in meeting these requirements. This paper deals with the first of these subdivisions.

The heating and ventilation of ships were originally carried on independently. Fresh air was distributed throughout the ships by a system of ducts with outlets in the various compartments, while heating was accomplished by radiators of steam coils mounted usually upon the bulkheads. This procedure required many radiators with extensive steam piping, all paralleled with necessary drain lines to return the condensate to the engineers' water tanks. The purpose of the drain pipes was to conserve fresh water which, of course, is quite important aboard ship. The existence of numerous valves and joints where leakage and waste of fresh water could occur was therefore objectionable.

The distribution of cold air throughout the ship also was objectionable because it prevented the uniform heating of spaces. Air velocities used for ship ventilation are higher than employed elsewhere, and the projecting of a jet of cold air into a space would be discomforting to those required to remain in its path. The duct terminals were usually adjustable, having the equivalent of a universal joint for permitting the jet to be turned in any direction; but at best the arrangement was unsatisfactory. Another objection to the independent heating and ventilating of ships was that such an arrangement did not readily permit of humidity control, for it was not always practicable to introduce moisture into the air with the system.

Introduction of Combined Heating and Ventilating System

The combined heating and ventilating system was introduced about 1908, the Delaware, North Dakota, Utah and Florida being the first ships on which this system was used. The first arrangement consisted practically in placing steam coils in the air ducts. The ducts were enlarged somewhat where the coils were placed, but the provision for heating consisted substantially of this simple arrangement. Gradually, the heating apparatus was developed until its present form consists of a so-called *thermo-tank* containing steam coils, placed in the ventilation trunk near the blower through which the air is blown before distribution throughout the space supplied by that blower. By this procedure, practically all of the disadvantages of the independent method are eliminated.

This arrangement, however, is not employed exclusively throughout the ship. There are certain spaces, which by reason of their location or ventilation arrangements, must still be heated by radiators. These spaces include the chart house and other adjacent spaces in the superstructure of the ship and a few of the bathrooms.

There are other spaces in the ship which are heated by a combination of direct radiators and heated air. Examples of these spaces are the captain's cabin, stateroom, and bathroom, where heated air is supplied, but where, by reason of the exposed location, such heating alone is insufficient. Fin-type steam radiators of compact construction are used to make up the deficiency in heating.

Finally, there are spaces aboard ship where ventilation is necessary, but where no heat is required. The magazines, for example, are supplied with unheated air. Certain of the storerooms also come within this category. For such compartments separate ventilation systems are needed which distribute air directly as taken from the outdoor supply trunk without sending it through a heater box, or else divert a portion of the delivery from a blower without sending it through a heater box. The latter arrangement is usually employed for storerooms, while magazines may have independent cold air systems.

Description of Ventilation System

Before stating the requirements for ventilation systems, both as governed by good practice, and as specified in the ship specifications, a general description of the ventilation methods is desirable. As has already been stated, fresh air is distributed throughout the ship by trunks and ducts which lead from intakes in the upper part of the ship to blowers and thence to the various points of delivery. It is of course necessary to provide for the escape of air from the various compartments within the hull to which air is supplied with the result that there are actually two general sets of ventilation systems within the ship, namely, supply and exhaust. The supply is normally from the blowers, that is, a so-called mechanical supply, while the escape of air is effected through the slight positive pressure produced in the spaces, such arrangement for escape of air being designated for natural exhaust. In certain spaces such as the evaporator rooms, ventilation is

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accomplished both by mechanical supply and mechanical exhaust. Certain compartments of this nature in which it would not be desirable to set up conditions by mechanical supply whereby air would tend to escape from these compartments into adjoining spaces through doors and other openings, are ventilated by natural supply and mechanical exhaust. These spaces are therefore under a slight negative pressure and the tendency is for air to enter them rather than to escape from them. Examples are toilet spaces and battery charging rooms.

The blowers used for ship ventilation range in general from capacities of 500 cfm to 10,000 cfm, the latter, for example, being of the propeller type for the ventilation of an evaporator room where large volumes of air are required in order to keep the temperature in the space down to an endurable point. The machinery spaces constitute special cases and require 25,000-cfm blowers for each of the engine rooms. The compartments included in the system supplied by any one blower are ordinarily grouped vertically in order to keep to a minimum the perforation of transverse bulkheads by ventilation ducts. This specification as to the grouping of the compartments into systems and the designation of blower capacities is usually indicated on the contract plans or type plans furnished by the government to the shipbuilder at the time of award of the contract.

Description of Heating System

Heating, as previously stated, is accomplished by installing heater boxes in the ventilation ducts just beyond the blower. In certain cases, it has been found practicable to conserve deck space by mounting the blower directly on the heater box. The admission of steam to the heater coils is governed by a thermostat mounted at what is expected to be a representative point in the various spaces served by that heater. This method of temperature regulation works satisfactorily for heaters supplying open spaces such as crew's quarters. However, where the heater supplies a number of closed spaces such as offices or staterooms, it is difficult to find a location for the thermostat bulb that will maintain a temperature from the heater satisfactory for all the spaces. One solution of this difficulty will be given later in this paper.

A rather broad method of temperature regulation is also provided. This method involves an adjustable by-pass in the heater box which permits a portion of the cold air to go directly through the box without passing over the heater coils. This by-pass feature is obviously employed in warm weather when it is not desirable to include the resistance of the heater coils into the system.

Heating and Ventilating Requirements

Definite temperatures are maintained in various spaces. These range from 65 F in shops to 70 F in living spaces and 75 F in hospital spaces. Restrictions are placed upon the velocities of discharge from terminals. In quarters and living spaces, the air velocities at terminals are not permitted to exceed 1,000 fpm. In other spaces the delivery velocity may reach 1,500 fpm. Natural exhaust ducts must be so proportioned that when they are carrying away the rated supply of air to a space, the velocity will be about 1,000 fpm.

Where ducts pass through ballistic or armor decks, the ballistic resistance of the deck must be maintained by fitting an armor grating in the opening. However, it is possible to avoid the use of such a grating where the duct size will permit by the use of a Venturi, with throat at the deck not exceeding 5 in. in diameter. This restriction makes it possible to conduct an 8-in. mechanical supply duct or a 10-in. natural exhaust through an armored deck without requiring an armor grating.

All ventilation piping must be so installed as not to impair the watertightness of the vessel. This necessitates such thickness of material and character of construction that the systems can resist the water tests placed on the compartment structure, or specifically, ventilation ducts in the hold of the vessel must be capable of withstanding water pressures amounting to a head of perhaps 25 ft. Furthermore, fittings or watertight closing covers must be provided for shutting off the ventilation duct in each watertight compartment. This requirement generally results in the installation of distributing piping within the space of light or non-watertight characteristics and having portable sections, removable to permit fitting the watertight cover if necessary to isolate the space.

It should be noted that for firerooms having two forced draft blowers, a special feature is necessary on the air intakes to the blowers. Airtight shutters must be provided which will close when subjected to pressure from below, or in other words, act as a check valve. This feature is necessary to prevent loss of air pressure to the oil burners under a boiler in the event of failure of one of the blowers. Without such shutters, the pressure from the blower which continues to operate would be lost by short circuiting through the disabled blower.

The fans are universally driven by electric motors. The power of the motors range from ½ hp for the smallest sets up to 15 hp for the 25,000-cfm fans for engine room ventilation. The motors are required to be adjustable as to speed and rated for continuous duty. They are either shunt or compound wound, and the speed adjustment must be of such character that the motors may operate continuously at minimum speed without exceeding allowable temperature rises. For fans driven by motors in excess of 2 hp, the controllers must be of the magnetic contactor type with speed adjustment above and below the rated speed of the motor, the range of adjustment being 15 per cent above and 15 per cent below rated speed. The adjustment for speed above rated speed is provided to cover the contingency that the resistance of the system be found too high after installation to obtain rated delivery without increase of speed. The controllers are also required to be self-starting upon return of voltage at any speed position of the controller.

Both centrifugal and propeller or disc types of fans are used. The centrifugal fans are standardized

for the following capacities in cubic feet per minute: 300, 500, 1,000, 2,000, 3,000, 4,000, 5,000, 6,000, 8,000, 10,000, 12,000, and 20,000, these capacities being the volumetric delivery of standard air at a static pressure of 2 in. of water at velocities ranging from 1,500 fpm for the smallest fan to 4,000 fpm for the largest size, through piping ranging from 6 in. in diameter for the smallest fan to 30 in. in diameter for the largest fan.

The fans are ordinarily required to be built with casings of steel plate or of aluminum alloy plate. For fans up to 2,000 cfm size, cast-iron or cast-aluminum shells may be used. All casings must be so constructed that the fans may be used for either clockwise or counter-clockwise rotation, and for eight angles of discharge, 45 deg apart, one of which must be horizontal.

All ventilating sets are required to pass complete tests at the manufacturers' works to determine that the fans have the specified capacity and that the sets have the required mechanical and overall efficiency. The data obtained from fan tests must, of course, be reduced to terms of standard air. The details of fan test procedure are fully covered in the Navy Department Specification for fans.

In the case of propeller-type fans, certain stipulations are made as to mechanical features, namely, the fans and motor shall be suitable for either horizontal or vertical operation and with the shaft vertical, the fan shall be either above or below the motor as required. For centrifugal fans, a mechanical efficiency of at least 50 per cent is required. For propeller fans the corresponding figure is 40 per cent.

The heater boxes and ventilation ducts and branches are required to be lagged, except that ducts and branches need not be lagged in spaces in which they supply hot air. The lagging for ducts consists of two layers each ¼ in. thick of corrugated asbestos paper covered with 10 oz canvas. In determining the performance of ventilating systems containing heaters, allowance is made for the resistance of the heater by specifying that the system shall deliver not less than the rated volume of standard air with heater coils by-passed, and not less than 70 per cent of the rated volume of standard air when the air is passing through the heater coils, but further, that under the latter condition the resistance of the heater shall not exceed one-half of the static pressure specified for the fan.

In proportioning heaters, it is required that the compartment temperatures shall be mintained when the temperature of the outside air is 0 deg F. It is undesirable to raise the temperature of the air as delivered at the terminals to too high a point. Although not specified, good practice limits this delivery temperature to approximately 120 F. Heating systems may be further required to have means for supplying sufficient moisture to the heated air to maintain a 50 per cent relative humidity at 70 F when the entering temperature is 0 deg F, and to have means for the automatic control of the humidity. The usual arrangement is a steam jet in the delivery side of the heater box. Such arrangements are not looked upon with favor by the ships' officers. Ships are operated on a competitive basis for economy in fuel use, and as the humidifiers require the use of steam jets with entire loss of the fresh water consumed therein and as all fresh water must be distilled aboard ships from sea water, the engineer officer is inclined to look upon the humidifiers in terms of the fresh water they consume, or rather, in terms of the fuel oil required for its distillation.

In certain cases where the propelling machinery uses superheated steam and where no saturated steam line has been available, a special problem has come up in the selection of the material for the heater coils. These would normally be of a copper composition, namely, a so-called admiralty metal. Copper alloys are particularly susceptible in their physical characteristics to the influence of elevated temperatures, and the temperature of the superheated steam is above the critical temperature of the alloy. For such situations, steel tubes would be required for the steam coils of the heaters. The steel tubes, generally, are objectionable because of their liability to corrosion. The obvious solution is to use saturated steam obtained preferably from a desuperheater rather than from an independent saturated steam line from the boiler steam drum.

As previously stated, thermostatic control of heaters is unsatisfactory where the system supplies a number of individual spaces such as staterooms. The most recent development of heating systems avoids such objection by distributing cold air to the staterooms, with small individual heater boxes mounted directly in the discharge terminals. This arrangement resembles certain modern types of heaters developed for heating larger spaces ashore, such as factories, warehouses, etc. The individual heater boxes are provided with adjustable vanes for directing the air stream from the terminal and also with a by-pass to permit a portion of the cold air to discharge directly from the terminal, and with an independent damper in the duct ahead of the heater. The arrangement, however, has one of the objections that were raised to the former use of individual steam radiators, namely, the extensive steam and drain piping required for the numerous individual heaters.

There are certain installations for which there are special requirements. One of these is that certain of the spaces aboard ship may be required to be maintained tenable as to temperature and respirability of the air when such spaces are completely sealed off from connection with the outside air. To meet such conditions a combined air purifying and cooling system is installed to recirculate the air within the space. The ventilating fan draws air from a system of suction terminals distributed through the space and delivers the air to discharge terminals after passing through a purifying chamber and over cooling coils, thereby maintaining a circulation through the space. The cooling coils, through which sea water circulates, lower the temperature of the air and also condense the excess vapor. The purifying chamber contains trays of a chemical preparation which absorbs carbon dioxide from the air. As oxygen is consumed by the personnel within the space, it is restored from a supply of compressed gas maintained for that purpose. In designing the cooling system, all sources of heat within the space must be considered. The radiation of heat from the electrical apparatus is readily determined. It is interesting to note that the radiation from the personnel within the space may amount to a considerable proportion of the total. For purposes of estimating

it is considered that this source of heat amounts to 120 watts per hour per man.

Similarly, in storage battery rooms it is necessary to provide sufficient air to carry off the heat developed in the charging and use of the batteries as well as the hydrogen gas evolved in charging. A storage battery is rapidly damaged if allowed to reach even a moderately warm temperature. The ventilation requirements for this purpose are based on an allowance of 0.06 cfm of air per cell per ampere normal charging rate or 0.15 cfm of air per cell per ampere finishing rate of charge for the batteries. If rheostats or charging panels are included, an additional supply of air amounting to 0.21 cfm per watt of electrical loss or 3.7 cfm per minute per Btu of heat evolved in the space, is required.

Spaces in which gasoline is carried require special care. Mechanical exhaust is used, the fan is required to have a brass impeller to avoid sparks, and a special flame arrester is included in the ventilation duct to prevent an explosion from striking back down the duct. This arrester consists of a number of thin, closely-spaced metal sheets arranged edgewise to the air stream. Its action depends upon the same principle as the action of the gauze in the miner's safety lamp.

Finally, it should be emphasized that dependence is placed upon ventilation for cooling and controlling the temperatures of the magazines where explosives are carried. The ventilation provisions must be such as to limit the magazine temperatures to a range of from 90 F to 110 F, depending upon the character of the explosive when the outside air is at a temperature of 80 F.

Insulation

All of the foregoing deals with the supply of heat and fresh air to the various compartments of a vessel. An equally important consideration is that of heat insulation. All heated spaces require insulation, not only to aid in retaining heat within the space, but also to prevent the condensation of moisture, where exposed metal surfaces may be chilled below the dew-point temperature. The latter possibility arises with pipes and ducts carrying cold air and water, as well as with exposed portions of steel structure subjected to cold outdoor temperatures. For water pipes, hair felt insulation is usually employed. For cold air ducts, a so-called cork paint is used.

The surface is painted with an adhesive upon which granular cork is thrown while the adhesive is still tacky. There results a loosely aggregated layer of cork approximately ¼ in. thick which, after further protection by paint, effectively prevents condensation for ordinary conditions.

The living spaces in the vessel require insulation, not only to retain heat in cold weather, but also to exclude heat in the summer and in the tropics. The steel decks and sides of the vessel may reach a surprisingly high temperature when exposed to the full intensity of the sun. The wood deck planking aids in screening from such heat, but interior insulation must be employed where decks are of bare steel and upon the sides of the vessel. Furthermore, the enclosures which surround the uptakes from the boilers to the smokepipes are subjected to high temperatures and must be covered to protect the personnel from such source of heat. The magazines in the hold are exposed not only to the machinery spaces, *i.e.*, firerooms and engine rooms as sources of heat, but, in addition, they are usually adjacent to fuel oil tanks. The latter contain steam coils for heating, for unless heated, the viscosity of the oil is too high to permit pumping. Accordingly, the magazines must be insulated.

Gravity Convectors

A heating unit, if placed within an enclosure, is a convector. The unit may be placed in (1) a cabinet located within the room, (2) a flue in a wall adjacent to the room or (3) a casing or enclosure located exterior to the room. There are, however, various modifications of these arrangements. An inlet for the cooled air and an outlet for the heated air must necessarily be provided. If the heating unit is located in the basement, ducts leading to and from the enclosure must also be provided. If the air flow over the unit takes place by natural convection, it is termed a *gravity convector,* whereas if the air flow is produced by a fan or blower, it is termed a *fan convector.*

It will be evident that a gravity convection system is essentially an indirect system of heating in which the air is recirculated from the floor of the room and delivered back again through the outlets at the top of the flue or cabinet at a temperature sufficiently great so that in cooling to the inlet temperature it gives up enough heat to supply the heat loss from the room. An exactly similar process goes on in a gravity circulating warm-air furnace heating system in which the air is returned to the furnace from the house. While gravity warm air heating could be included in this classification, the principles involved are entirely different.

An enclosed heating unit containing steam or hot water (convector) emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible. A small amount of heat is transmitted by radiation to the inside surface of the enclosure, the amount so transmitted diminishing as the surface temperature of the enclosure approaches the surface temperature of the heating unit. A partially shielded unit located within the space to be heated will transmit heat by radiation to the objects in the space it can *see.*

The general effect of an enclosure placed about a heating unit of the type used as a direct radiator is to decrease the amount of steam that the unit will condense per hour and thus to reduce the amount of heat given off by the unit. The results of recent investigations indicate that only where the enclosure is made much higher than the heating unit, thereby producing a chimney effect, is the condensation rate equal to or greater than that of an identical radiator unenclosed.

NOMINATIONS FOR 1932

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1932, takes pleasure in submitting the following list of nominees:

For President:

F. B. ROWLEY, Minneapolis, Minn.

For First Vice-President:

W. T. JONES, Boston, Mass.

For Second Vice-President:

C. V. HAYNES, New York, N. Y.

For Treasurer:

F. D. MENSING, Philadelphia, Pa.

For Members of the Council:

Three-Year Term

F. E. GIESECKE, College Station, Texas.
G. L. LARSON, Madison, Wis.
J. F. McINTIRE, Detroit, Mich.
W. E. STARK, Cleveland, Ohio

Respectfully submitted,

NOMINATING COMMITTEE

R. C. BOLSINGER, *Chairman*

Chapters	*Representative*
Cleveland	H. M. NOBIS
Illinois	H. M. HART
Kansas City	N. W. DOWNES
Massachusetts	DAVID MOULTON
Michigan	J. F. McINTIRE
Minnesota	M. S. WUNDERLICH
New York	RUSSELL DONNELLY
Western New York	F. H. BURKE
Ontario	H. J. CHURCH
Pacific Northwest	E. O. EASTWOOD
Philadelphia	R. C. BOLSINGER
Pittsburgh	F. C. McINTOSH
St. Louis	C. A. PICKETT
Southern California	R. L. GIFFORD
Wisconsin	E. A. JONES

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. 9—*Section* 2. The Nominating Committee shall consist of a member designated by each Chapter, which shall effect its own organization and elect its own Chairman.

Section 3. The Committee may meet at the call of the Chairman, for discussion, but the official meeting shall be held at the Semi-Annual Meeting of the Society, at which time the candidates for each of the offices of the Society to be filled at the next Annual Meeting shall be selected by a majority vote of the members of the Committee present and voting. Five shall constitute a quorum.

Section 4. The Committee shall first secure the consent of all candidates selected, and shall, if possible, announce the names of the candidates at the Semi-Annual Meeting, but in any case the names of the candidates shall be certified to the Secretary of the Society at least four months before the next Annual Meeting. The Secretary shall publish these names in the October issue of THE JOURNAL.

Art. 10.—*Section* 3. The Secretary shall prepare ballots with the names of all candidates and forward them to the members at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election*. The election is governed by the By-Laws for the election of officers, with the single exception, that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the government of the Research Laboratory, adopted at the January, 1919, Meeting, and revised January, 1930, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1932:

D. E. FRENCH, Philadelphia, Pa.
F. E. GIESECKE, College Station, Texas
A. P. KRATZ, Urbana, Ill.
G. L. LARSON, Madison, Wis.
L. A. HARDING, Buffalo, N. Y.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

SECTION I—ORGANIZATION

Research Committee

1. There shall be a standing committee known as the Research Committee, consisting of fifteen members, each serving for three years, and five retiring each year.

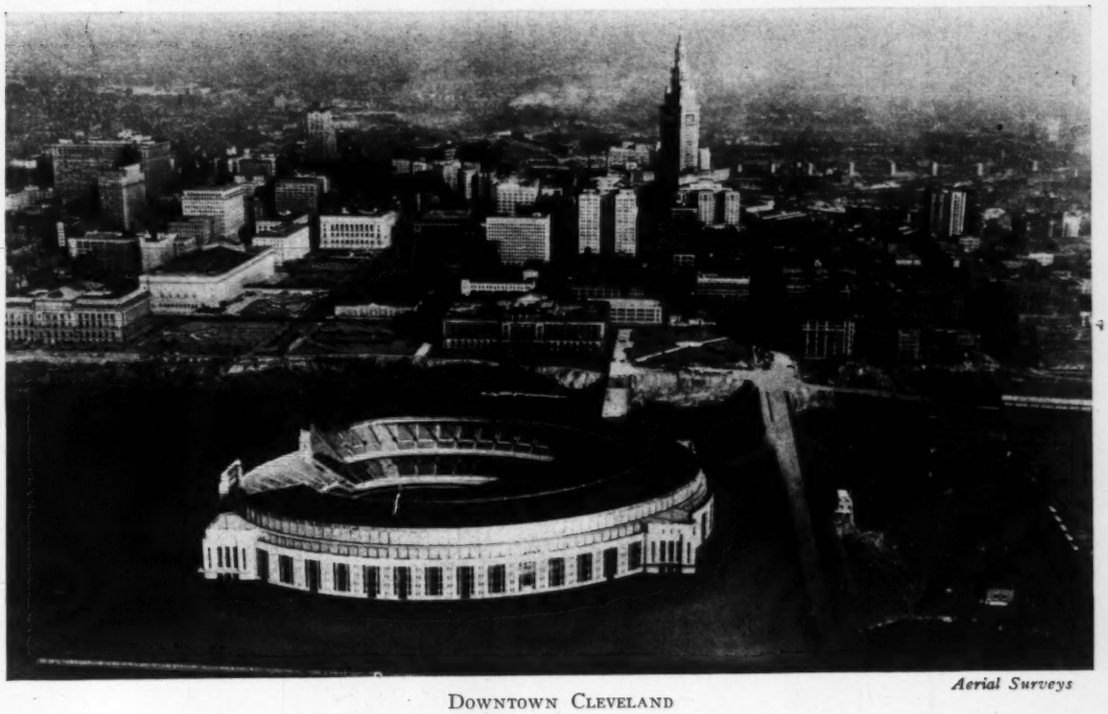
a. The Council shall nominate previous to October first of each year five members to fill the vacancies of those retiring at the next Annual Meeting.

b. The nominations made by the Council shall be published in the October issue of the Society's JOURNAL.

c. Prior to December first of any year, any ten members, over their own signatures, may nominate one or more additional members of the Research Committee, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

d. The election shall otherwise conform to the regulations provided for the election of officers of the Society.

e. Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.



Aerial Surveys

DOWNTOWN CLEVELAND

38th Annual Meeting at Cleveland

January 25-29, 1932

THE attention of the heating and ventilating and refrigerating professions is being focused on Cleveland, O., the scene of the respective Annual Meetings of the American Society of Heating and Ventilating Engineers and the *American Society of Refrigerating Engineers,* which convene during the week of January 25, 1932, and simultaneously the Second International Heating and Ventilating Exposition will be held at the Cleveland Auditorium. Members and guests from all parts of the world expect to find these events stimulating and instructive, and their plans are now being made to attend.

The opening session of the 38th Annual Meeting of the A. S. H. V. E. will be preceded on Monday, January 25, by meetings of the Society's Council and various Committees. The first technical session will be called to order by Pres. W. H. Carrier on Tuesday morning, January 26, in the ballroom of the Hotel Statler and from then on sessions will be held each day until Friday the 29th.

A feature of the meeting will be two joint sessions with the members of the *American Society of Refrigerating Engineers,* whose 27th Annual Meeting will be held at the Hotel Cleveland. At 10 A. M. on Wednesday, January 27, a joint session of the two Societies will be held at the Cleveland Auditorium, the program of technical papers being sponsored by the A. S. H. V. E. On the following morning, Thursday, January 28, at the Cleveland Auditorium, a second joint session will be held and the program is being arranged by the *A. S. R. E.*

Several joint session functions will be held commencing with an informal reception and dance on Tuesday evening, January 26, at the Hotel Statler. Several special events for ladies will be jointly sponsored.

The annual banquet of the *A. S. R. E.* is to be held on Wednesday, January 27, and the dinner and dance of the A. S. H. V. E. at the Hotel Statler will be held on Thursday evening, January 28, with W. C. Kammerer in charge.

Because of its large and representative number of manufacturing plants Cleveland offers many attractions for those who wish to inspect the industrial plants of the city. F. H. Morris is chairman of the Committee on Transportation and Inspection Trips and anticipates making a visit to the Goodyear hangar in Akron where the U. S. Navy dirigible *Akron* is housed.

Provision will be made to see various plants and an afternoon has been set aside for inspection trips. Each member will indicate his choice by advance registration.

Reception and registration of the members and guests will commence on Monday, January 25, and will be in charge of C. F. Eveleth. The Entertainment program is being directed by R. G. Davis, and Publicity is being prepared by M. F. Rather. The chairman of the Finance Committee is Walter Klie.

The general chairman of arrangements is T. A. Weager, and acting for the *A. S. R. E.* in coordinating the various functions is D. F. Keith.

CLEVELAND MUSEUM OF ART

HOTEL STATLER,
A. S. H. V. E. HEADQUARTERS

38th ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

January 25-29, 1932
Hotel Statler, Cleveland, Ohio

Monday—January 25th

10:00 A. M.—Meeting of the Council
12:30 P. M.—Luncheon for Officers and Authors
2:00 P. M.—Meeting Committee on Research
7:00 P. M.—Joint Dinner-Meeting of the Councils of A. S. H. V. E. and *A. S. R. E.* and their wives (Hotel Cleveland)

Tuesday—January 26th

10:00 A. M.—First Session (Ballroom Hotel Statler)
Address of Welcome
Greeting by Pres. W. H. Carrier
Report of Tellers of Election
Report of Council
Technical Papers (*two*)
Report of the Finance Committee
Report of the Secretary
2:30 P. M.—Report of Committee on Ventilation Standards
Discussion—Revision of Constitution and By-Laws
7:00 P. M.—Informal Reception and Dance for A. S. H. V. E. and *A. S. R. E.* members (Hotel Statler)

Wednesday—January 27th

10:00 A. M.—Third Session (Cleveland Auditorium)
Joint Meeting with *A. S. R. E.*—Program arranged by A. S. H. V. E.—Pres. W. H. Carrier presiding
2:30 P. M.—Inspection of Second Annual Heating and Ventilating Exposition
7:00 P. M.—Dinner for Past-Presidents and Wives (Hotel Statler)

Thursday—January 28th

10:00 A. M.—Fourth Session (Cleveland Auditorium)
Joint Session with *A. S. R. E.*—Program arranged by *A. S. R. E.*—Glen Muffly presiding
2:30 P. M.—Fifth Session
Technical Papers (*three*)
7:00 P. M.—Annual Banquet and Dance (Hotel Statler)

Friday—January 29th

10:00 A. M.—Sixth Session (Hotel Statler)
Technical Papers (*two*)
Installation of Officers
Resolutions of Officers
Adjournment
12:30 P. M.—Luncheon and Meeting of Council
2:30 P. M.—Inspection Trip to Cleveland Industrial Plants

COMMITTEES

T. A. Weager, *General Chairman*

Finance: Walter Klie, *chairman;* W. E. Stark and H. M. Nobis.
Banquet: W. C. Kammerer, *chairman;* C. W. St. Clair, Vincent Eaton.
Reception and Registration: C. F. Eveleth, *chairman;* F. A. Kitchen, C. Gottwald.
Transportation and Inspection Trips: F. H. Morris, *chairman;* C. J. Deex, J. E. Beyer.
Entertainment: R. G. Davis, *chairman;* E. H. Pogalies, S. H. Givelber.
Publicity: M. F. Rather, *chairman;* D. E. Humphrey.
A. S. R. E.: D. F. Keith.

Local Chapter Reports

Kansas City

The opening meeting of the Kansas City Chapter for the 1931-32 season was held at the Kansas City Club, September 8, 1931 with 30 members in attendance, and Vice-President W. A. Russell in the chair.

A brief resume of the technical and entertainment programs of the Society's Semi-Annual Meeting in Swampscott was given by E. K. Campbell, who also explained the value to be derived from attendance at national meetings.

Announcement was made of the next Annual Meeting of the Society in Cleveland, Ohio and Vice-President Russell gave assurance that Kansas City would be well represented.

A few facts about current Society activities were given by A. V. Hutchinson, Secretary of the Society. It was brought out that additional members were being added to the Society rolls each month and payment of dues for the first eight months of 1931 was on exactly the same ratio as for the corresponding period of the previous year. The members present indicated their interest in the Research Program of the Society and the efforts of the Council to assist Chapters with their local meeting programs.

The members indicated their appreciation for the information given about current Society affairs and Chairman Russell expressed the wish of the Chapter members that Pres. J. M. Arthur would soon be with them again.

Massachusetts

The monthly meetings of the Massachusetts Chapter of the Society were held from October 1930 to May 1931, inclusive, and a greater attendance was recorded at noon meetings than those held in the evening.

The opening meeting of the season was held at the new Edison Steam Service Station, Boston, where the Chapter members, together with the Boston Real Estate Exchange, were the guests of the Boston Edison Co. With competent guides the 150 members and guests made a tour of inspection through the entire plant, and later enjoyed the dinner which was served on one of the upper levels of the premises. After the dinner, Pres. T. F. McCoy opened the meeting and then turned it over to D. S. Boyden, who acted as toastmaster, introducing I. E. Moultrop, chief engineer, Edison Co. After Mr. Moultrop welcomed the Society members and guests, several of the Edison engineers spoke briefly on some phase of the distribution of steam.

The topics which were discussed at the remainder of the meetings and the speakers are as follows:

November, 1930—J. A. McCarthy: Representative of the Baker Supply Co. Discussed the application of special fittings to welded pipe lines. Henry Dusenberry: Progress in Manufacture and Performance of Steam Traps.

December, 1930—Pres. L. A. Harding: Utilization of the Sun's Energy. C. W. Farrar: Increase of Membership. A. V. Hutchinson: Society Affairs.

January, 1931—Alfred Kellogg: Manufacture and use of Pottery in Trade.

February, 1931—D. C. Lindsay: Modern Trends in Air Conditioning.

March, 1931—R. B. Wilson: Statistical Data on General Business Conditions.

The present officers of the Massachusetts Chapter are as follows:

President—David Moulton.

Secretary-Treasurer—Leslie Clough.

Board of Governors—David Moulton, R. T. Kern, Leslie Clough, R. F. Gifford, C. R. Swaney and F. R. Ellis.

Pacific Northwest

The Pacific Northwest Chapter of the Society has just completed the most successful year in its history under the leadership of Prof. E. O. Eastwood.

At the first fall meeting, held in October 1930, the Chapter was fortunate in having A. V. Hutchinson, Secretary of the Society, as one of the guests. Mr. Hutchinson gave an interesting talk on the Society's activities. This was followed by entertainment which was provided by J. Rowley and Miss L. Jones of the State game and preservation, showing moving pictures of the scenery and wild life of the State.

One of the largest meetings of the season was held in December, at which J. D. Ross, superintendent of the City Light, delivered a lecture and a practical demonstration of the transmission of electricity without a conductor. Mr. Albue, engineer of the Liquid Carbonic Co., also spoke at this meeting, demonstrating the use and manufacture of dry ice, or condensed CO_2. Mr. Albue's talk was followed by a talk on the use of gas for heating and the future of the gas industry in the Northwest, which was given by Mr. Pollard, manager of the Seattle Gas Co.

F. C. Houghten, Director of the Society's Research Laboratory, was the guest at the February meeting. Mr. Houghten delivered a very interesting and detailed report of the activities of the research laboratory at Pittsburgh, illustrating his talk with slides.

At the March meeting R. S. Whaley, president of the Power Plant Engineering Co., spoke on the combustion of oil and the practical application of oil burners to heating plants, which was followed by a report by Lyle Dudley on the progress of the proposed law for professional engineers.

The April meeting, which was the largest of the season, had as its principal speaker, W. H. Carrier, president of the Society. President Carrier delivered an address on Servicing the Human Power Plant, which was well illustrated with slides. A. V. Hutchinson, secretary of the Society, was also one of the guests at this meeting.

Prof. E. O. Eastwood spoke on aviation at the May meeting, illustrating his talk with slides.

The following officers have been selected for the year 1932:

President—W. E. Beggs.

Vice-President—C. F. Twist.

Treasurer—W. W. Cox.

Secretary—M. Anderson.

Board of Governors—E. O. Eastwood, E. L. Weber and L. W. Dudley.

New York

The opening meeting of the 1931-32 season will be held at the Building Trades Club, October 19, and a unique program is in prospect, according to an announcement by Pres. Russell Donnelly. Members will have the opportunity of meeting the new officers, and several entertaining topics of discussion have been provided by the chairman of the Program Committee, M. W. Ehrlich.

Arrangements have been made to have a joint meeting November 16, with the metropolitan section of the *A. S. M. E.*, and the subject to be presented is, Studies of Corrosion in Heating Systems.

Chapters

THE 1931-32 season of Chapter activities begins during the current month when the local associations start their monthly meetings. Subjects of local and special interest are being provided by the Chapter Officers for the instruction and entertainment of the Society members.

The opening meetings of Pittsburgh, Cleveland, Michigan, Wisconsin and St. Louis Chapters will have Pres. W. H. Carrier as the featured speaker.

A new project, being sponsored by the Council of the Society to assist local Chapter Officers in obtaining outside speakers, has been started under the direction of a special Council committee whose personnel is: R. H. Carpenter, *Chairman,* E. K. Campbell and Roswell Farnham.

For many months this Committee has been studying Chapter meeting programs and will soon be prepared to suggest subjects for discussion and furnish a list of available speakers. The plan, as outlined by Mr. Carpenter, provides for the establishment of a clearing house at the headquarters office of the Society

Cleveland

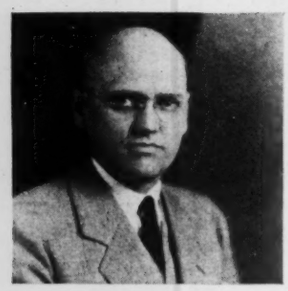
R. G. DAVIS

Illinois

A. G. SUTCLIFFE

C. W. DELAND

Kansas City



J. M. ARTHUR

D. D. ZINK

Massachusetts

DAVID MOULTON

LESLIE CLOUGH

Michigan

G. H. GIGUERE

G. D. WINANS

Minnesota

H. E. GERRISH

A. B. ALGREN

New York

RUSSELL DONNELLY

W. A. SWAIN

where Chapters may secure cooperation of the Committee in the arrangement of monthly meeting programs. Subjects of vital interest that may appeal to several Chapters can through cooperative planning be presented to several Chapters by the same speaker. This will tend to conserve the time of prominent men, save unnecessary expense and provide the best available talent.

In the belief that the members will be interested in knowing some of the officers who are serving the Chapters during the 1931-32 season, their photos are being presented herewith.

From time to time during the year special announcements, from the Society Officers and Council will be presented by these Chapter Officers on subjects of widespread interest.

Philadelphia

E. N. SANBERN

W. P. CULBERT

Pittsburgh

F. C. HOUGHTEN

J. L. BLACKSHAW

St. Louis

R. M. ROSEBROUGH

C. R. DAVIS

Southern California

O. W. OTT

H. B. KEELING

Wisconsin

J. G. SHODRON

H. F. HAUPT

Western New York

JOSEPH DAVIS

J. J. LANDERS

Ontario

H. S. MOORE

H. R. ROTH

Pacific Northwest

W. E. BEGGS

M. ANDERSON

Exposition in Cleveland To Present New Equipment

January 25, 1932

WHEN the Second International Heating and Ventilating Exposition opens at the Cleveland Auditorium, January 25, 1932 simultaneously with the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the 27th Annual Meeting of the *American Society of Refrigerating Engineers*, visitors will find that many products that have been in the Laboratory and test rooms of manufacturers will make their first public appearance. With public interest concentrated on air conditioning and cooling for the home and other living quarters, automatic fuel burning and temperature control, visiting engineers will find much to attract them. Exhibitors will use equipment in full operation to demonstrate its effectiveness in providing improved methods of heating and ventilating for comfort.

In the warm-air furnace section, it is significant that new designs tend toward a wider participation in air conditioning by filtering, humidifying, heating and cooling the air for both summer and winter service.

New boiler and radiator designs will be revealed and of special interest are the new types of combination units, using gas, oil or coal.

The gas heating division will give striking illustration of progress made in both domestic and industrial heating.

In the oil burner section many new designs will reflect the improved standards of service that are characteristic of present day types of automatic units.

A domestic stoker division will be a prominent feature as well as the larger types used for heavy duty work in schools and other public buildings.

Refrigeration for air cooling has made rapid strides and some unique applications will be demonstrated by leading manufacturers.

Fans, unit heaters, ventilators, automatic temperature controls and valves and specialties in infinite numbers will attract the interest of visiting engineers.

Various organizations, which include the *American Gas Association, American Oil Burner Association, National Warm Air Heating Association, Heating and Piping Contractors National Association, American Society of Refrigerating Engineers* and the A. S. H. V. E., will have appropriate booths.

Cleveland will be the Capitol of the Heating and Ventilating World in January.

In the list of exhibitors will be found the following leading concerns:

Adams Bros. Mfg. Co., Inc.
Aerofin Corp.
Aerologist, The
Air Reduction Sales Co.
American Air Filter Co.
American Gas Association
American Gas Products Co.
American Oil Burner Assn.
American Propeller Co., The
American Radiator Corp.
American Society of Heating and Ventilating Engineers
American Society of Refrigerating Engineers
American Temperature Indicating Co.
Ames Iron Works
Ames Pump Co., Inc.
Armstrong Machine Works
Armstrong Cork Co.
A. G. A. Testing Laboratory
Automatic Burner Corp.

B-Line Boiler Co.
Barber-Colman Co.
Barber Gas Burner Co.
Barnes & Jones
Bishop & Babcock Sales Co.
Bonney Forge & Tool Works
Brownell Co., The
Bryant Heater & Mfg. Co.
Buckeye Blower Co.

Carrier Corp.
Carrier Engineering Corp.
Carrier-Lyle Corp.
Century Electric Co.
Century Engineering Corp.
Chicago Pump Co.
Cleveland Gas Burner & Appliance Co.
Cleveland Steel Products Corp.
Cook Electric Co.
Cooling and Air Conditioning Corp.
Coppus Engineering Corp.
Crane Co.
Crystal Oil Burner Corp.

Davison Gas Burner & Welding Co., N. C.
Domestic Engineering
Dunham Co., C. A.
Struthers Dunn, Inc.

East Ohio Gas Co., The
Economy Pumping Machinery Co.
Emerson Electric Mfg. Co.

Engineering Publications, Inc.
Estate Stove Co., The

Fitzgibbons Boiler Co., Inc.
Fox Furnace Co.
Frick Co., Inc.
Fuel Oil Journal

Gas-Aire, Inc.
General Electric Co.
General Gas & Light Co.
Grinnell Co., Inc.

Hart Oil Burner Corp.
Heating Journals, Inc.
Heating, Piping & Air Conditioning
Heating & Ventilating
Henry Furnace & Foundry Co., The
Hill Co., E. Vernon
Hoffman Specialty Co., Inc.

Illinois Engineering Co.
Iron Fireman Mfg. Co.
Independent Air Filter Co.

Johnson & Co., S. T.
Johnson Service Co.
Johns-Manville Corp.

Kewanee Boiler Corp.
Kieley & Mueller, Inc.

Kleenaire Filter Co.

Leeds & Northrup Co.
Linde Air Products Co.

McCord Radiator & Mfg. Co.
McDonnell & Miller
Marsh & Co., Jas. P.
May Oil Burner Corp.
Mercoid Corp.
Mineral Felt Insulating Co.
Minneapolis-Honeywell Regulator Co.
Monarch Mfg. Co.
Motor Stoker Corp.
Mouat Vapor Heating Co.
Mueller Furnace Co., L. J.
Multicell Radiator Corp.

Nash Engineering Co.
National Radiator Corp.
Nelson Corp., Herman
Niagara Blower Co.

Oil Heat
Oxweld Acetylene Corp.
Oil Heating Institute

CLEVELAND AUDITORIUM

Owens-Illinois Glass Co.

Parker Appliance Co.
Patrol Valve Co., The
Peerless Unit Ventilation Co.
Penn Electric Switch Co.
Pennsylvania Furnace & Iron Co.
Pierce Butler & Pierce Mfg. Co.
Piqua Electric Mfg. Co.
Plumbers Trade Journal Publishing Co., The
Powers Regulator Co.
Preferred Utilities Mfg. Corp.
Propellair, Inc.

Rawlplug Co., Inc., The
Raymond Co., F. I.
Reif-Rexoil Co., Inc.
Richardson & Boynton Co.
Robertson Co., H. H.
Rochester Circulator Sales Co.
Ruud Manufacturing Co.
Russell & Co., W. A.

Sanitary & Heating Age
Sarco Co., Inc.
Silent Automatic Corp.
Silent Glow Oil Burner Corp.
Spencer Heater Co.
Stay-Rite Co., Inc.
Sun Radiator Cover Co., Inc.
Superior Steel Corp.
Surface Combustion Co.
Swartwout Co., The

Taco Heaters, Inc.
Time-O-Stat Controls Co.
Trerice Co., H. O.

Union Carbide Co.
United States Radiator Corp.

Warm Air Heating
Warren Webster & Co.
Warm Air Furnace Fan Co.
Webster Tallmadge Fan Co.
Weil-McLain Co.
Whittington, Inc., W. P.

York Heating & Ventilating Corp.
York Ice Machinery Corp.
Young Radiator Co.
Youngstown Sheet & Tube Co.

President's Home Building Conference in Washington

ON September 15, President Herbert Hoover announced that the Conference on Home Building and Home Ownership will be held in Washington D. C. December 2 to 5, when a report of a nation wide study of housing problems will be presented by 25 committees, composed of 400 representative people interested in building and housing activities.

The conference has been organized under the direction of Secretary Thomas P. Lamont and the discussion in December will embrace finance, design, equipment, city planning, household management and many other aspects of home construction ownership and environment.

Among the special fields of study is that of Committee 2 on Fundamental Equipment which includes heating, lighting, plumbing, ventilation and air conditioning. Sub-division 1 of Committee on Fundamental Equipment includes the following members who met September 3, at Atlantic City, N. J.: Prof. A. C. Willard, *Chairman,* Urbana, Ill.; W. H. Driscoll, New York; Philip Drinker, Boston, Mass.; and Perry West, Newark, N. J.

The Committee on Fundamental Equipment consisting of 15 members is headed by Prof. C. P. Bliss of New York University.

The 25 committees now functioning and their chairmen are as follows:

Types of Dwellings—John Ihlder, Pittsburgh.
Fundamental Equipment—Prof. Collins P. Bliss, of New York University.
Kitchens and Other Work Centers—Miss Abby L. Marlatt, University of Wisconsin.
Utilities for Houses—Morris Knowles, Pittsburgh.
Subdivision Layout—Harland Bartholomew, St. Louis.
Business and Housing—Harry A. Wheeler, Chicago.
Industrial Decentralization and Housing—Stuart W. Cramer, Cramerton, N. C.
Blighted Areas and Slums—Abram Garfield, Cleveland.
Reconditioning, Remodeling and Modernizing—Frederick M. Feiker, Washington.
Construction—A. P. Greensfelder, St. Louis.
Design—William Stanley Parker, Boston.
City Planning and Zoning—Frederic A. Delano, Washington.
Finance—Frederick H. Ecker, New York.
Taxation—Dr. T. S. Adams, of Yale University.
Home Ownership and Leasing—Ernest T. Triggs, Philadelphia.
Home Furnishing and Decoration—Miss Ruth Lyle Sparks, New York.
Landscape Planning and Planting—Mrs. Junius S. Morgan, Princeton, N. J.
Household Management—Miss Effie Raitt, of the University of Washington.
Housing and the Community—Dr. Joseph Hersey Pratt, Boston.
Farm and Village Housing—Provost A. R. Mann, of Cornell University.
Negro Housing—Nannie H. Burrlighs, Washington.
Home Information Centers—Miss Pearl Chase, chairman, Santa Barbara, California.
Homemaking—Miss Martha Van Rensselaer, of Cornell University.
Large-Scale Operations—Alfred K. Stern, Chicago.
Relationship of Income and the Home—Professor Niles Carpenter, of the University of Buffalo.

Six committees have been named to correlate the work of the separate groups, as follows:

Standards and Objectives—Lawrence Veiller, of New York.
Research—Professor James Ford, of Harvard University.
Legislation and Administration—Bernard J. Newman, Philadelphia.
Education and Service—Dr. Albert Shaw, New York.
Organization Programs—Local and national, Miss Harlean James, Washington.
Technological Developments—Dr. George K. Burgess, Director of the Bureau of Standards.

Members to Nominate Candidates for Anderson Medal

THE Committee on Award of the F. Paul Anderson Medal extends an invitation to all members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, to submit the names of candidates for the first award of the F. Paul Anderson Medal, to be made at the 38th Annual Meeting of the Society January, 1932.

The conditions are briefly as follows:

1. Any member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in good standing during 1931 is eligible for nomination..
2. Each nomination sent to the Committee on Award must be accompanied by a statement of the candidates accomplishments specifically mentioning the work done or services performed and indicating in what way such work or services have been outstanding in the field of heating, ventilating or air conditioning, during any year or years prior to 1931.
3. All nominations must be in the hands of the Committee on *November* 30, 1931, and are to be addressed: Committee on Award of the F. Paul Anderson Medal, c/o Secretary, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 51 Madison Ave., New York, N. Y.

The personnel of the Committee on Award is as follows: Prof. A. C. Willard, Urbana, Chairman; Thornton Lewis, Newark, N. J.; L. A. Harding, Buffalo; F. B. Rowley, Minneapolis; W. T. Jones, Boston and F. C. McIntosh, Pittsburgh.

President Carrier to Address Chapters

During the month of October Pres. W. H. Carrier will speak to five local chapters. On October 5, Mr. Carrier will meet with the Pittsburgh members and on the 7th Cleveland Chapter will entertain him.

The first meeting of the Michigan Chapter October 12 will be featured by Mr. Carrier's address on Ventilation Requirements and Standards and on October 13 and 14, Mr. Carrier will appear before the Wisconsin and St. Louis Chapter members, speaking on Servicing the Human Power Plant.

In November, on the same day that the Council will hold a meeting in Buffalo, Mr. Carrier will speak before the Western New York Chapter members and on November 10, the Ontario Chapter members will hear him talk on the Economic Viewpoint in Air Conditioning for Human Comfort.

Macy S. Good Division Manager for Dunham

Macy S. Good, after 10 years as manager of the Chicago sales office of the C. A. Dunham Co., commenced his duties as Division Manager, on September 1. Mr. Good, who was graduated from Purdue University in the class of '99, will have his headquarters in the Dunham Bldg., Chicago.

Through his activities in the Illinois Chapter of the Society, the Architects' Club of Chicago, and his prominence in Rotary, his genial personality will not be lost to the large number of friends he has among the architects, consulting engineers and heating contractors of that city.

S. E. Dibble at Patton School for Boys

The new superintendent of the Patton School for Boys at Elizabethtown, Pa., is S. E. Dibble, past-president of the Society and formerly Theodore Ahrens' Professor at Carnegie Institute of Technology. Professor Dibble continues in the work of training young men in which he has been engaged for the past 16 years at this institution for boys founded by Thomas Ranken Patton.

Research Director Addresses A. P. H. A.

At the annual convention of the *American Public Health Association* held in Montreal, F. C. Houghten, Director of the Society's Research Laboratory in Pittsburgh, addressed the Public Health Engineering Section September 17, on the subject of, Heat and Moisture Dissipation from the Bodies of Children and Its Application to School Ventilation. This address prompted an extensive discussion and the health officials in attendance expressed satisfaction with the scientific investigation of this problem.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month, 6 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by October 25, 1931, these candidates will be balloted upon by Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Gross, Lyman Charles, Chief Mech. Engr., Magney & Tusler, Inc., Minneapolis, Minn.	D. M. Forfar H. J. Sperzel	G. R. Magney W. F. Uhl
Knee, J. Stuart, President, Knee Heating Co., Grand Rapids, Mich.	W. W. Bradfield B. A. Parks—(*A.S.M.E.*)	W. J. Carroll Charles Morton—*Non-Member*
Meisel, Carl Louis, Sales Engr., The F. M. Webber Co., New York, N. Y.	Dr. A. A. Adler C. L. Meyer	F. W. Moore—(*A.S.M.E.*) E. F. Church—(*A.S.M.E.*)
Morton, Harold S., Ch. Engr. and Genl. Mgr., Stott Stoker Co., St. Paul, Minn.	Albert Buenger H. J. Sperzel	A. B. Algren J. V. Martenis
Wierenga, Peter O., Vice-Pres. & Supt., Cox James Co., Grand Rapids, Mich.	W. W. Bradfield	W. J. Carroll
Ziesse, Karl L., Secy. & Treas., Phoenix Sprinkler & Heating Co., Grand Rapids, Mich.	W. J. Carroll W. W. Bradfield	C. H. Alexander

Heating · Piping and Air Conditioning EDITORIALS

Proper Piping Saves Money

Excessive friction in a piping system due to improper size or lay-out is just as expensive as is excessive friction in a machine, though probably in most cases it is much less noticeable. There are potential power savings in the majority of industrial plants that can be had through a careful check of the piping and the making of changes where they are found advisable.

Frequently, sharp turns in a piping system introduce unnecessary resistance to flow; it is often possible to make the necessary changes at small expense compared with the economies that will be had. The pipe sizes and the friction head should be checked; if the latter proves unduly large, the installation of larger pipe may be indicated. Excessive vibration of a pump may denote sharp bends or obstructions in its suction line, causing turbulence and preventing a full free stream of the liquid.

It would seem helpful to look on any piping system as a transportation system. Excessive friction interferes with the transporting of the fluid—steam, water, air, oil gas, brine, process liquids, etc.—and represents a real money loss in the power required. Though friction in piping is more difficult to determine than friction in a machine, for instance, its presence should not be dismissed from the mind of the piping or maintenance engineer.

Instruments Cut Costs

Accurate day-to-day and month-to-month records of fuel or steam consumption used for heating will indicate sources of waste and are of inestimable value to the engineer in charge of the mechanical equipment in an industrial plant, office building or other structure. Proper instruments for making the collection of the necessary data both possible and practicable are soon paid for in reduced operating costs.

There are many plants not now equipped with adequate facilities for measuring steam or fuel consumption. When it is realized that with the proper records waste of heat can be reduced to the minimum, it is uneconomical to withhold the small capital investment in the right instruments for obtaining the proper information to enable the keeping of such records.

Progress must be based on knowledge. Progress in revising heating costs downward depends on a thorough knowledge of what they are and on what they depend.

Proper Installation

The last few years have seen rapid developments in insulation for piping and other purposes; the maintenance of quality has constantly been stressed and upheld by manufacturers. No less important than the proper insulating material is its correct application to the job.

That the manufacturers of insulation and the contractors applying it realize the great importance of the proper installation of insulation is indicated by the organization of bureaus which give a bond to the owner or his agent certifying that the insulation will meet the specifications of the engineer or architect.

Particularly in the heating, piping and air conditioning fields, the proper installation of equipment is of equal importance with its proper design and manufacture. Engineers should lay great stress on the necessity of installing machines and materials correctly.

Electricity for Heating

It is understood that several unit fabricated steel buildings will be constructed in Chicago late this year or early next year. Among the structures proposed are an office building, an apartment house and a residential development. It is planned to heat the office building—which will be air conditioned, of course—by electricity. According to *Steel,* the exterior walls will be of 18-gage stainless steel or aluminum, carefully insulated. Heat loss will be cut down by utilizing double panes of plate glass separated by a one-inch air space. Ac-

cording to estimates and preliminary tests the cost of heating will not be out of line with buildings constructed along more customary lines.

Building heating by electricity is a development being watched carefully by many heating engineers at the present time. A large office building in California has been completed recently which is heated by the reversed refrigeration cycle, supplemented by small direct heaters. Much of the same equipment is used for cooling in summer.

It is believed that electricity will continue to find numerous applications where its use for space heating will be indicated. Undoubtedly, the Chicago development will give the heating engineer much helpful data.

Reading Technical Journals—A Step Toward Success

The other day I received a hard luck letter from an engineer. He said his financial condition was desperate. He had taken a temporary job shoveling coal. I have known this man for years. I recall certain things about him. I recall that he had access to several engineering journals, but he never read them. He told me so.

I cannot help but think there is some connection between these two facts. We have heard, of course, many times, the classical statement "Tell me how you spend your spare time and I will tell your fortune."

Everyone has—or should have—spare time. What is done with it is extremely important. It may not seem so right now, but the accumulation of misused spare time is a liability that increases with the years.

I do not know of a successful engineer who does not keep up-to-date by reading current technical literature. There is an art in reading technical journals; its cultivation and practice will aid success.

—An Engineer.

Practical Piping Problems

Piping Maintenance in Freezing Weather

W. H. Wilson's Monthly Discussion of the Problems He Meets From Day-to-Day

Freezing weather increases the responsibilities of the maintenance engineer. Fig. 1 shows pipe and fittings from water lines that have been damaged by freezing.

Gate valves, under some conditions, are easily damaged by freezing, especially when the valve is closed tight. A small amount of water that is left between the gates and the valve body may leave the valve body as shown in Fig. 2, upon freezing.

An incident is recalled where a large size iron body gate valve was given a water pressure test in the shop. The valve was left closed and taken to the job. Not being immediately required for installation, it was left on the floor of the building near an outside doorway. That night the outside temperature went below the freezing point and drafts of cold air struck the valve. The freezing of the small amount of water that was left between the gate and the valve body resulted in a broken body casting, together with added expense in replacement and delay to the installation work.

Attention to Check Valves

Freezing has a tendency to lift the bonnets of globe valves and the caps of check valves. Check valves should be given close attention when draining and blowing out pipe lines preparatory to protection from freezing weather. Check valves may be placed so as to hold back a portion of the water and moisture in the pipe lines. It is advisable to remove the interior discs and gates for complete protection.

Globe valves are likely to retain water in the pocket caused by the shape of the interior construction of the valve body. Fig. 3 shows a cap of a check valve that was lifted by freezing.

Gate Valve Damaged

Fig. 4 shows a gate valve located on the riser pipe of a water supply to a building and a shut-off valve of the stop and waste pattern on the horizontal underground line outside of the building. When this piping was

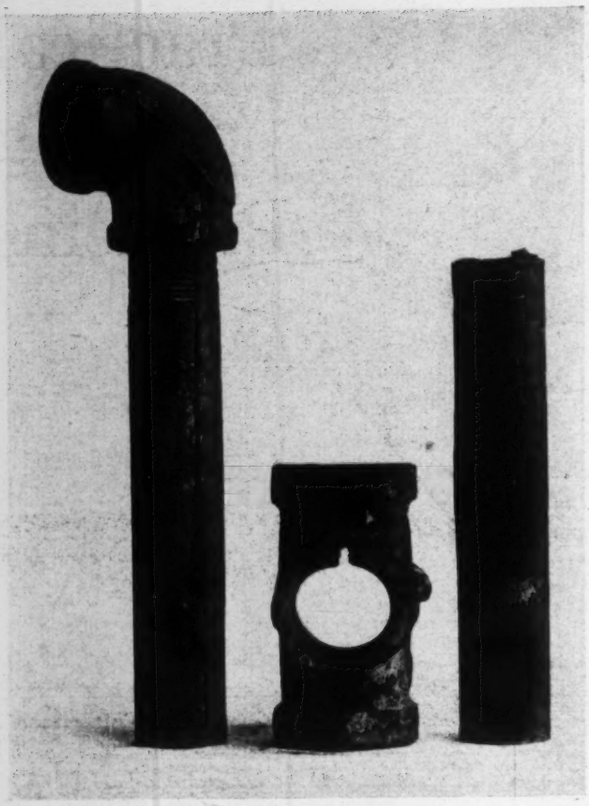


FIG. 1—PIPE AND FITTINGS FROM WATER LINES THAT WERE DAMAGED BY FREEZING

drained and blown out before freezing weather, the pipe fitter closed the gate valve on the riser pipe, drilled and tapped an opening in the body of the valve above the gate, attached a hose and blew the piping out with compressed air. He then closed the underground stop and waste valve outside of the building, which shut off the water and automatically drained the piping up to the gate valve on the riser pipe. Through an oversight this gate valve was left closed. When freezing weather arrived the body of the gate valve was damaged by the small amount of water that was left in the space between the gate and the valve body.

Special attention should be given to valves when draining and blowing out pipe lines. If the bonnet of the valve is in a horizontal or downward position, water will collect in the lower part of the bonnet and the valve may be damaged by freezing.

Valves equipped with dash-pots and diaphragm chambers should be thoroughly drained of water and moisture when not in use during the cold weather. A small amount of water left in these valves will freeze and damage expensive equipment. One of the best ways to obtain complete protection is to take the valves apart and remove the dash-pot pistons.

Use of Anti-freeze Solution

Hydraulic operating valves require very close attention when water is used in the hydraulic equipment. The spindles, pistons, packing leathers and other features make the draining and blowing out of such equipment a problem. It is usually a good plan to remove the spindles from the operating valves and loosen the packing box glands in order that the moisture in the water-soaked packing will not freeze and strain the gland bolts. It is difficult, in many cases, to remove all of the water by draining and blowing out. It is advisable to use an anti-freeze solution as an additional protection.

Shut-off Valves Must Be Tight

One trouble that is often experienced in protecting piping from damage by freezing is the failure of shut-off valves to hold tight. A small amount of water leaking through a valve continuously will fill a pipe line and equipment that has been previously drained and blown out. A case is recalled where a large steam supply line leading from one building to another, a distance of several hundred feet, and installed overhead on supports outdoors, was shut off and drained. The shut-off valve

FIG. 2—VALVE BODY DAMAGED WHEN A SMALL AMOUNT OF WATER FROZE

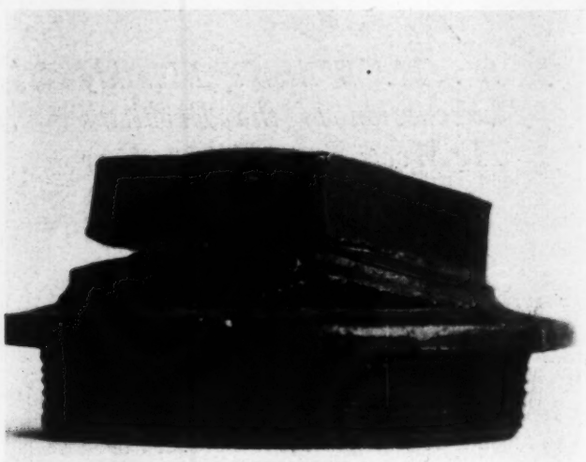


FIG. 3—CAP OF A CHECK VALVE LIFTED BY FREEZING

leaked slightly and the drain valve was closed, for some unknown reason, after the line was drained. This allowed water to accumulate in the large pipe line and freeze during the cold weather. When the steam supply line was needed, which happened in the middle of the heating season, it was necessary to thaw it out by using electricity. In cases of this kind it is advisable to separate the flanges on the outlet side of the shut-off valve and insert a blind gasket, or wedge apart the flange on the pipe line from the valve. By using a steel plate for a blind gasket, it is possible to put a compressed air pressure test on the piping system, to determine whether there are any leaks in the return lines,

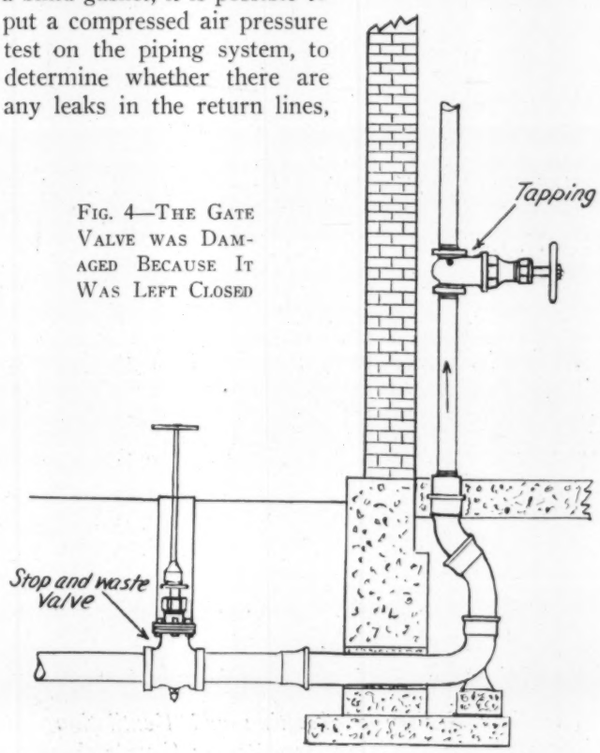
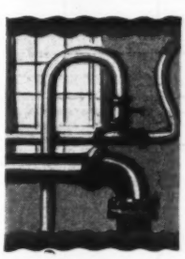


FIG. 4—THE GATE VALVE WAS DAMAGED BECAUSE IT WAS LEFT CLOSED

broken radiators or other defects, before steam is turned on and the equipment put in regular use.

Watch These Points

The proper protection of extensive piping systems requires close attention and careful inspection of all the equipment. Special care should be given to pipe lines, radiators and heating coils located under roofs, in skylights, back of stairways, in elevator pent houses, heating coils in tanks, etc. A radiator trap that is blocked with pipe scale or out of order will allow the heating units to fill up with water and freezing weather may cause damage. Long return lines in skylights or under floors may have been installed so as to allow pockets or the hangers and supports may have been broken, allowing water to accumulate.



Proper Pipe Line Support

Extracts from the minutes of a weekly discussion held by a large operating construction company. Meetings are held for educational purposes as well as to create a closer cooperative feeling between employer and employe.

Fred (The Foreman): "Well, Henry, what have you to offer today relative to that problem you worked on over at the Ideal Mfg. Co. plant?"

Henry: "You know, fellows, we can't be too careful these days when we put in pipe lines. It may be okay for some shops to have their men jump in and cut loose without much thought or supervision, but, not so with us. We have a reputation to sustain.

"There was that outfit that installed all the main high pressure water and air lines on this big job that I am to talk about, taking it away from our folks at approximately 14 per cent less than we bid, so the boss told me.

"Here is what I found; I hate to say it but it almost served the Ideal people right. The main water line through the plant (used for fire and other purposes) was 8-in. extra heavy galvanized pipe, with gas welded joints. Ordinarily this should seem to be about all that could be expected for they went to the extra trouble of having the pipe coated with some kind of a bituminous wrapped material to guard against soil corrosion which is a good thing, as you know. A sort of an insurance measure, the manager remarked, for they knew the line was to run through a heavy fill that had all kinds of rubbish in it.

"This is what happened: About one-fourth way up the main shop under the concrete floor and close to a heavy boring mill foundation (that vibrated quite a bit when operating) one of the welded joints gave way. The floor had cracked and sunk a little, probably due to the heavy castings laying on it and at this point water came through like Niagara when the leak occurred.

"The line was about 4 ft. 6 in. below floor level and when we uncovered the pipe I saw the joint was parted on the bottom side about a half inch (see Fig. 1.) This was too much to try and fill in with a torch and I figured the line must have dropped down somehow so I had it stripped for over 50 feet each way and then put a straight edge on it; the line had settled over 3 inches and in so doing opened up at the joint."

"What would cause a joint like that to give way?" chimed in the boss. "We are supposed to make joints as strong as the pipe or even stronger. You boys all know that your examination test joint had to stand a very rigid test before you were permitted to use a torch in the field."

Henry: "The engineers were on hand to check up and see what the trouble was and on an analysis of some drillings taken from the broken joint it was found to have been made from a poor grade of welding rod. Also, the workmanship was not so good for they noted several cold spots in the joint and from the nature of the fracture (ragged tear) it was thought that no attempt had been made to normalize and keep the strains and stresses spread.

"I only emphasize this point so you will always be

careful when making joints that will be covered up; I know it was a real lesson to me. However, here is where the real trouble was. Let me have the chalk, boss, and I will show you on the blackboard what was the basic cause of the break. (See Fig. 2.)

"The pipe came in from the street main through the concrete foundation where an opening had been provided and rested on the bottom of the hole. From there it ran

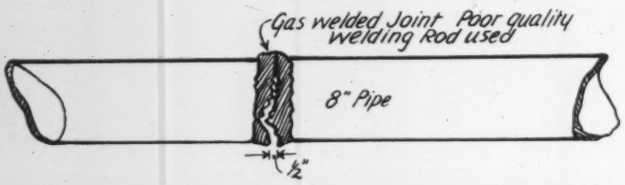


FIG. 1—THIS SKETCH ILLUSTRATES HOW THE JOINT WAS BROKEN. THE RAGGED SECTION POINTS TO POSSIBLE LACK OF NORMALIZING

straight some 120 feet to a concrete trench, resting there, also, on the bottom of a cored hole.

"You can see two solid resting points and 120 feet of pipe resting on a fill which evidently settled when heavy castings were placed on the floor and that, combined with vibrations from the boring mill that I spoke of before, caused the pipe to travel downward.

"Before we got through the job I recommended three concrete piers under the pipe at points shown. The plant engineer said okay and agreed that if proper thought had been given the installation in the beginning by the original contractor, that his people would have been glad to allow an extra for the installation of piers or other support.

"The specifications did not provide any, and the 'hurry up and get done boys' did not use their heads as they might have. With proper support there is not much doubt that the joint would have held.

"This was a long story, boss, but I think it was justified in more ways than one for, aside from the value of checking up on our own practice, it was a costly lesson to the Ideal folks. I understand the repairing of the leak alone cost them $875.00 in real money, not to say anything about a lot of embarrassment and delay incidental to the water being shut off and also having two motors (that were located in pits below floor level) blow up.

"All this happened because the mechanic in charge of the work failed to support properly or recognize the necessity of supporting his pipe. Of course, some fault can be charged to the joint and some to the designing engineer. But, I personally like to think this would not have occurred on any of our work. At any rate it provides for the opportunity of checking up and saying we should always use our heads as well as our hands."

Fred: "That is fine, Henry, and I know the boys will profit by this expensive lesson of the other fellow."

—*T. M. D.*

Conventions and Expositions

American Gas Association: Annual convention and exhibition, Oct. 12-13; Municipal Auditorium, Atlantic City, N. J.

Southern Power and Machinery Show: October 19-24; Textile Hall, Greenville, South Carolina.

National Association of Practical Refrigerating Engineers: Annual convention and exposition, Nov. 10-13; Municipal Auditorium, Houston, Texas. Secretary, Edward H. Fox, 435 N. Waller Ave., Chicago.

Third International Conference on Bituminous Coal: November 16-21; Carnegie Institute of Technology, Pittsburgh, Pa. Chairman, Thomas S. Baker, President, Carnegie Institute of Technology.

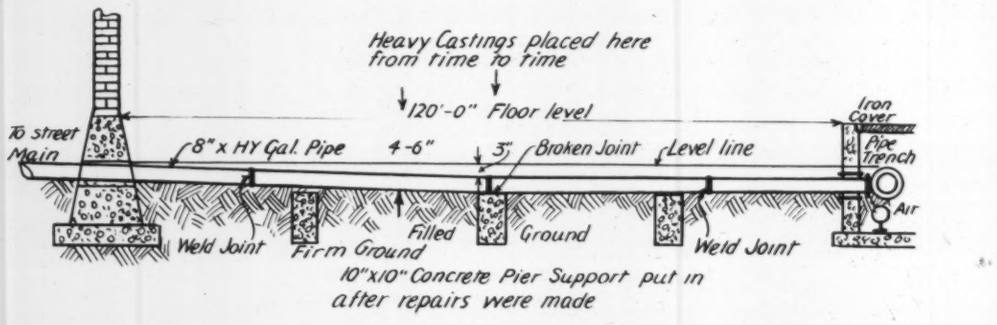
American Society of Mechanical Engineers: Annual meeting, Nov. 30-Dec. 4; New York City. Secretary, Calvin W. Rice, 33 W. 39th st., New York City.

American Society of Heating and Ventilating Engineers: Annual meeting, Jan. 25-28; Cleveland, Ohio. (Joint meeting with A. S. R. E.) Secretary, A. V. Hutchinson, 51 Madison ave., New York City.

American Society of Refrigerating Engineers: Annual convention, Jan. 26-29 (Joint meeting with A. S. H. V. E.); Hotel Cleveland, Cleveland, Ohio. Secretary, David L. Fiske, 37 W. 39th St., New York City.

International Heating and Ventilating Exposition: Jan. 25-29; Cleveland Auditorium, Cleveland, Ohio. Manager, Charles F. Roth, International Exposition Co., Grand Central Palace, New York City.

FIG. 2—REPAIRS WERE MADE BY CUTTING OUT 12 IN. OF PIPE AND WELDING IN A NEW PIECE

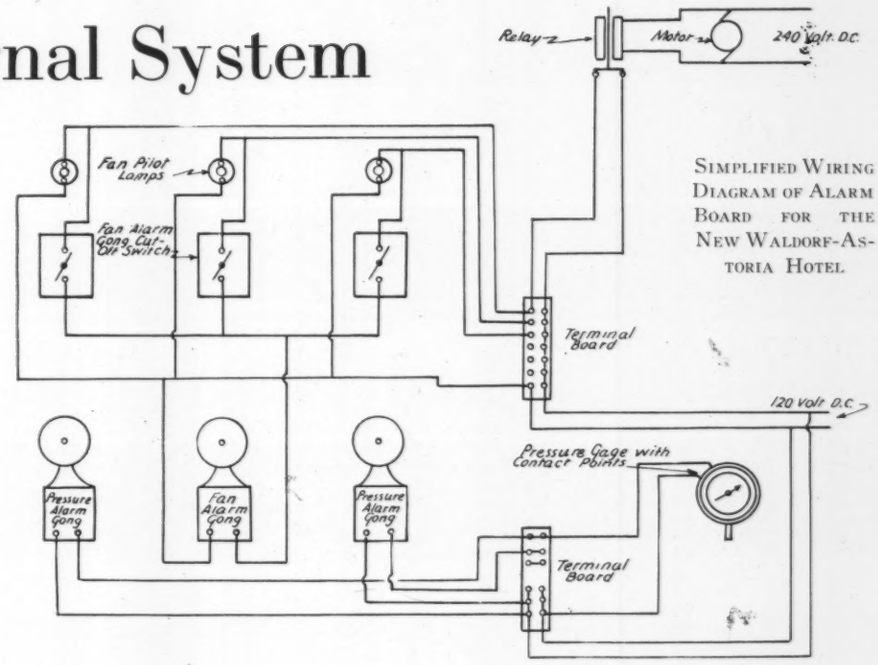


Proper Operation of Hotel Heating and Air Conditioning Equipment Insured by

A Signal System

By the Staff of
Clyde R. Place*

(Edited by Ernest Williams)



SIMPLIFIED WIRING DIAGRAM OF ALARM BOARD FOR THE NEW WALDORF-ASTORIA HOTEL

A TOTAL of over one million cubic feet of air per minute is required for the ventilating and air conditioning systems in the new Waldorf-Astoria Hotel in New York City. The movement of this large quantity of air requires a total of sixty-five fans.

With a system of such magnitude, it would be next to impossible for the engineer in charge to be certain at all times that the proper fans were in operation, and had not been stopped for some reason unknown to him. The same would also apply to the pressure reducing valves on the steam lines to the ventilating equipment, although in this case a hazard would be introduced due to the possibility of excessive pressures. For this reason, a signal system was designed to warn the engineer of the stopping of any fan, or of a pressure in excess of the setting of the alarm system on the low pressure side of any reducing valve above the ground floor.

As designed, the signal system for the fans consists of an individual pilot light for each fan and a common gong for all fans. A snap switch is also provided for each fan so that the gong may be turned off from that particular fan without affecting the pilot light. A relay mounted on each fan controller is connected in parallel with the fan motor after the circuit breaker and controller. The relay coils are wound for 240 volts, direct current, but the relay contacts and other equipment are designed for use with 120 volts, direct current.

The action of this portion of the alarm system is as follows: Current passing through the relay coil maintains an open circuit in the line to the pilot light and gong. Should the motor be stopped, a spring returns the relay armature to the closed-circuit position, actuating the gong and lighting the pilot. The engineer's attention is thereby called to the stopping of the fan and he may then turn off the gong, leaving the light burning. The pilot light and gong are connected in parallel to allow the gong to operate even though the lights may be burned out. It should be remembered that a ventilating system in a building of the type of the Waldorf-Astoria is required to function the majority of twenty-four hours; it was felt that greater economy would be obtained, and fewer renewals of lamps would be required, if they were lighted only when the fans were not running.

Alarm System for Reducing Valves

The alarm system for the pressure reducing valves, of which there are five, consists of a gage with electrical contacts located on the low pressure side of each reducing valve. An individual gong operated by 110 volts direct current is provided for each valve and is so wired that the gong will ring when the pressure exceeds that for which the contacts are set. No switches are provided for these gongs in order that they may be stopped only by remedying the trouble causing the alarm. No alarm is provided to indicate low pressure, since the pressures will be varied depending on the season of the year.

All alarm equipment is located on a separate panel below that containing the various pressure gages for the heating system. The board is of slate with black marine finish. All gages are flush mounted with chromium-plated mounting rings. The name plates are also chromium-plated but the switch plates have an oxidized nickel finish.

The entire signal system was designed with the end in view of providing maximum dependability and economy of operation and at the same time to warn the operating engineer of the stopping of any fans which might endanger the comfort of guests of the hotel.

* Consulting engineer, New York City.